JOURNAL

OF THE

AMERICAN SOCIETY OF NAVAL ENGINEERS

VOL. XXVIII. MAY, 1916. No. 2.

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U. S. S. *OKLAHOMA.*

DESCRIPTION.

BY HENDERSON B. GREGORY, ASSOCIATE.

The *Oklahoma*, battleship No. 37, is the last of our battleships, built or building, equipped with reciprocating engines. She is sister ship to the *Nevada*, fitted with Curtis turbines, and built by the Fore River Shipbuilding Corporation, of Quincy, Mass.

Authorized by an Act of Congress, approved March 4, 1911, the *Oklahoma* was built under contract by the New York Shipbuilding Company, of Camden, N. J., at a cost of $5,926,000. The time of completion, under the terms of the contract, was three years. She is a twin-screw vessel of about 27,500 tons displacement, and designed for a speed of 20.5 knots with the main engines developing 24,800 I.H.P. Her boilers are of the Babcock & Wilcox oil-burning type.

Principal Hull Dimensions.

Length between perpendiculars, feet and inches	575–0
on L.W.L., feet and inches	575–0
over all, feet and inches	583–0
Breadth, extreme, on L.W.L., feet and inches	95–2⅛
molded, feet and inches	94–10⅛
Depth molded, main deck at side M.S., feet and inches	44–5½
Draught, mean, to L.W.L., feet and inches	28–6
Displacement corresponding, tons	27,500
per inch at L.W.L., tons	93.6
Ratio of length to beam	6.04
Coefficient of fineness, block	0.617
midship section	0.984
L.W.L. plane	0.718

GENERAL DESCRIPTION OF HULL.

In profile the *Oklahoma* presents some striking departures from that of her predecessors, there being but one smoke pipe, four turrets and an upper deck carried aft about half the length of the vessel. Two masts of the customary cage type are provided and fitted with spotters' tops, wireless, signal yards, etc. There are eight searchlights, two on each mast and four mounted on portable trucks.

Superstructure, Deck and Bridges.—Forward between turret II and the foremast is located the superstructure deck. On it are mounted two 5-inch guns and four 3-pounders. Above the superstructure deck is the bridge provided with portable extensions reaching to the vessel's sides. The chart house is located amidship on the bridge, just forward of which is the conning tower. A conning platform is above the bridge amidship.

Upper Deck.—This deck extends from the bow aft about half the length of the vessel. Forward are the windlass, an electric deck winch and turrets I and II. It is weather deck throughout, except for deck house, just abaft turret II, which contains the captain's quarters. Aft of the deck house is the foremast, smoke pipe and stowage of boats.

Main Deck.—The main deck is a weather deck abaft the break in upper deck. Beginning forward are located the wardroom quarters, part of the 5-inch battery, galleys, bakery, butcher shop, etc. On the weather portion of the deck are stowage of boats, aftermast, two deck winches, turrets III and IV, and capstan aft.

Gun Deck.—Forward are located the wardroom, junior and warrant officers' quarters, abaft of which are crew's quarters, firemen's wash rooms, sick bay, offices, armory, post and printing offices, general mess condiment issuing room and pantry, amidship. Abreast of barbettes III and IV are chief petty officers' and crew's quarters, starboard and port, respectively, and crew's quarters aft containing laundry space and electric capstan. Part of the 5-inch battery is located in the crew's and chief petty officers' quarters aft.

COMPLEMENT.

The ship's complement will be approximately as follows:

Commanding officer,	1
Wardroom officers,	26
Junior officers,	16
Warrant officers,	12
Crew,	748
Marines,	61
Total,	864

BATTERY.

The main battery comprises ten 14-inch guns, arranged in four turrets along the center line of the vessel. Turrets I and IV mount three guns each, and II and III two guns each. Turrets I and II are grouped forward on the upper deck, the latter firing over the top of the former, and IV and III are similarly arranged on the main deck aft.

The secondary battery, of twenty-one 5-inch rapid-fire guns for torpedo defense, is divided starboard and port as follows: two on superstructure deck, twelve within inclosed portion of main deck and seven on gun deck aft, one of the latter being on center line astern.

The 5-inch battery is served by twenty-one electric-chain ammunition hoists, each driven by a 3-horsepower G. E. Co. motor.

The following smaller guns are also provided:

4 3-pounder guns for saluting;
2 1-pounder guns for boats;
2 3-inch field pieces;
2 0.30-caliber machine guns.

The torpedo equipment consists of four 6.8-meter by 21-inch submerged torpedo tubes.

SMALL BOATS CARRIED.

The following small boats constitute the ship's regular allowance:

2 50-foot steamers;
1 35-foot motor boat;
2 50-foot motor sailing launches;
2 40-foot motor sailing launches;
1 40-foot sailing launch;
1 31-foot racing cutter;
2 30-foot whale boats;
2 20-foot dinghies;
2 14-foot punts.

The 50-foot steamers are carried on the main deck, port and starboard, at break in upper deck, and the whale boats in davits at the ship's side abreast turrets III and IV. All other boats are stowed on the upper deck, port and starboard, abreast of smoke pipe.

Two electrically-operated boat cranes are provided for handling all boats, except the whale boats. Each crane has two operating gears, one for turning and the other for hoisting, driven by G. E. motors of 60 and 50 horsepower, respectively.

ANCHOR WINDLASS.

An electric-driven windlass, designed by the Hyde Windlass Co., of Bath, Maine, is located on the upper deck forward, with driving gear below on the upper platform.

The windlass is driven by two 150-horsepower G. E. motors, through worm gearing and shafting, and so arranged that the wildcats can be operated together or independently of each other.

DECK WINCHES.

Three deck winches are provided, one forward on the upper deck, abaft of the windlass, and two aft of the main deck, port and starboard, abreast pump-room skylight.

The winches are electric-driven, compound geared, and have a lifting capacity of 4,000 pounds at 200 feet per minute, or 16,000 pounds at 50 feet per minute, each. They are of the Hyde type, each driven by a 35-horsepower G. E. motor.

CAPSTAN.

A Hyde electric-driven capstan is located on the main deck, center line, abaft turret IV, with driving motor below on the gun deck. It is compound geared and driven by a 40-horsepower G. E. motor.

LAUNDRY.

The laundry is located on the gun deck aft, starboard side. It is equipped to wash for about 100 men and is provided with the following machinery, each machine driven by an independent motor:

No.	Type machine.	Type and H.P. of motor.
1	Washer,	Diehl, 1½
1	Extractor,	Diehl, 1½
1	Combination ironer,	Diehl, ½
1	Band ironer,	Diehl, ⅛
1	Reverse body ironer,	Diehl, ½

2 Washtubs;
1 Soap tank, 90 gallons;
1 Starch kettle, 10 gallons;
1 Steam-heated bosom press;
1 Ironing table;
1 Drying room (conveyor type).

GALLEY OUTFIT.

Officers' Galley.—The officers' galley is equipped with oil-burning range of four sections.

Crew's Galley.—In the crew's galley are installed the following:

1 oil-burning range of eight sections;
4 steam-jacketed copper kettles (80 gallons each);
2 steam-jacketed copper kettles (60 gallons each);
2 coffee urns, 100 gallons each;
1 electrically-driven meat grinder;
1 electrically-driven egg beater and cake mixer combined;
1 mechanically-driven meat slicer;
1 five-section warming oven.

Bakery.—The following equipment is installed in the bakery:

2 No. 1 U. S. Navy standard, electric bake ovens;
1 steam box;
3 portable dough troughs;
1 electrically-driven dough-mixing machine.

Miscellaneous equipment located as noted:

1 electrically-driven potato peeler, in potato-peeler room;
1 electrically-driven dish washer, capacity 6,000 pieces per hour, in general mess pantry;

1 electric butter slicer, in general mess issuing room;
1 bread slicer, in bread room;
1 20-gallon coffee urn, in evaporator room;
1 ice cream freezer.

FIRE MAIN.

The fire main is supplied by seven fire and bilge pumps, located in the pump rooms and firerooms.

The main is entirely below the protective deck and extends throughout the machinery spaces, close under the berth deck, in two 6-inch lines, port and starboard. It is cross-connected at the forward and after ends of the pump rooms and in the forward fireroom, and supplied by two 6-inch risers from the four main fire and bilge pumps, and three 5-inch risers from fireroom fire and bilge pumps, with cutout valves at the main. From the forward and after cross-connections, 6-inch branches extend in single line forward and aft of the machinery space, at the lower platform level, reducing in size at extreme ends to 4½ and 5 inches, respectively.

On the gun deck aft there is a 5-inch connection to the independent sanitary system for the crew's water closets and washroom. There is a stop valve where it joins the latter.

There are 5-inch connections to the sanitary system in the port engine room, and a 5-inch by-pass to same in the forward fireroom and on gun deck aft.

SANITARY SYSTEM.

The sanitary main is 5 inches in diameter, and is supplied by five 5-inch risers from the fire and bilge pumps, a 5-inch connection from the fire main in the port engine room, and two 5-inch by-passes, one forward and one aft, from the fire main.

The main extends in single line, port side, through the machinery space, rising forward and aft of same to the berth deck space, where it leads forward and aft close under the gun deck. Well aft it rises to the gun deck, where it joins the

crew's independent flushing system, with stop valve at junction.

Branches, as required, are led to the chief petty officers' washroom and water closets, sick bay, bath, laundry, general mess pantry, galleys, bakery, firemen's washrooms, junior, warrant and wardroom officers' lavatories and water closets, etc.

Aft there is an independent system for the exclusive use of the crew's washroom and water closets. This system is supplied by two motor-driven, direct-connected, centrifugal pumps, of about 500 gallons per minute capacity each. The main is 5 inches in diameter, with the necessary branches to the plumbing fixtures.

FRESH-WATER SYSTEM.

The following fresh-water tanks are provided:

Compartment or tank.	Location.	Between frames.	Gallons.
A-6-P	Hold, port	20-23	16,065
A-6-S	Hold, starboard	20-23	16,146
Mess attendants' washroom supply tank.	Gun deck, starb'd.	95-96	100
Chief petty officers' washroom supply tank.	Gun deck, starb'd.	115-116	150
Firemen's washroom supply tank	Main deck, port	64-65	250
Do	Main deck, stb'd	64-65	250
Battle-dressing station supply tank.	Berth deck, port	121-122	125
Do	Berth deck, stb'd	48-49	125
Laundry tank	Gun deck, stb'd	123-124	200

The hold tanks have two 2½-inch filling connections from the ship's sides forward, and also a 2½-inch connection from the distiller main. From these tanks water is pumped through the fresh-water main by means of two direct-connected electric centrifugal pumps of 200 gallons capacity per minute each, driven by G. E. Co. motors of 12 horsepower each. Branches are led as required, from the main to the various lavatories, pantries, etc., and the small gravity tanks enumerated above.

REFRIGERATING PLANT.

The ice-machine room is on the lower platform, port side, between the engine and fireroooms. The plant consists of two 3-ton horizontal, steam-driven Allen dense-air ice machines of the Navy type, fitted with recoolers, two 9-can Bureau standard ice-making boxes and the cold-storage rooms, located on the berth deck above the ice-machine room. The cold-storage rooms are divided into four compartments for meat, butter, crew and officers, the crew's room serving also as vestibule. The rooms are fitted with 1-inch galvanized-iron pipe coils, through which the cooling agent circulates. The supply and return air mains are of copper, 3 inches diameter, from which branches are taken through suitable valves or manifolds to the cold-storage rooms, ice tanks and scuttle butt brine-cooling tank, the scuttle butts being brine cooled.

RESERVE FEED TANKS AND CONNECTIONS.

Double-bottom compartments B-92 to B-95, inclusive, are fitted up as reserve feed tanks.

The tanks are filled through a 3½-inch pipe fitted with two 2½-inch hose connections at the ship's sides, and connected to the combined filling and suction manifold in fireroom No. 2.

There is also a 2½-inch filling pipe from the distiller fresh-water main, which is led to the same manifold. From the manifold 2½-inch combined filling and suction pipes are led to the bottom of each tank.

There is a 2½-inch connection between the manifold and the auxiliary-feed suction main, that the auxiliary-feed pumps may draw from the tanks; and the auxiliary-feed pump in fireroom No. 2 is arranged to draw direct from any tank and discharge to any other tank. The main feed pumps have a 2½-inch independent suction from the manifold, for taking water direct from any tank, and the usual ¾-inch make-up feed connection to the main condensers is provided.

VENTILATING SYSTEM.

Artificial ventilation is provided where necessary for all quarters, living spaces, passages, storerooms, magazines, engine rooms, dynamo rooms, evaporator room, etc. There are fifty-five ventilating fans and motors, each on its own circuit. In general air is supplied on the plenum system to the different compartments requiring ventilation. The toilet spaces and engine rooms are also provided with the exhaust system. The ventilating fans were made by the B. F. Sturtevant Co., Boston, Mass., and the motors by the G. E. Co.

Heater boxes are fitted in the ventilating ducts to all quarters below the main deck, crew's space, etc., for heating the incoming air in cold weather; no other heating apparatus being provided for these spaces.

HEATING SYSTEM.

All staterooms and quarters below the upper deck, crew's space, etc., are heated by the thermo-tank ventilating system; heater boxes provided with suitable steam coils being placed in the air duct for heating the air supplied to these compartments. The remaining portions of the vessel, except the turrets, which are provided with electric heaters, are heated by the customary pipe-coil steam radiators.

The heating system is divided into two main sections, one forward and one aft. The former takes steam from the main steam cross connection in fireroom No. 1, and the latter from the auxiliary steam loop in the pump rooms. Stop and reducing valves are fitted at the branches from the main and auxiliary steam pipes. The plant is further subdivided into fourteen circuits, as listed in Table I.

MAIN ENGINES.

There are two main engines designed to develop collectively 24,800 indicated horsepower when making 125 revolutions per minute. They are placed abreast in two separate water-

TABLE I

Circuit No.	Steam from	Drain to	Circuit supplies.
1	Forward heating system manifold.	Radiators to trap in engine room. Thermo-tanks to traps at each thermo-tank. Traps discharging to feed and filter tanks and condensers.	Radiators in Captain's quarters. Thermo-tanks, gun deck frame #55 and berth deck frame #57, starboard.
2			Thermo-tank, gun deck, frames #54-57, port.
3			Thermo-tank, gun deck, frames #72-75, port.
4			Thermo-tank, gun deck, frames #72-75, starboard.
5			Thermo-tank, main deck, frame #60, center line.
6	Forward galley manifold.	Galley, bakery, pantries and radiators to traps in engine room discharging to feed and filter tanks and condensers. Water heaters to auxiliary exhaust.	Officers' and crew's galley and bakery. Crew's W.C. midship. Water heaters in firemen's wash rooms.
7			Captain's bath heater and pantry. Warrant officer's bath and pantry. Wardroom and junior officer's pantries.
8			Water heaters and radiators in junior and wardroom officers' quarters. Executive officers and spare bath, and water heater in forward battle dressing station.
9	After heating system manifold.	Thermo-tanks to traps at each thermo-tank. Radiators, pantries and laundry to traps in engine room. Traps discharging to feed and filter tanks and condensers. Water heaters to auxiliary exhaust.	Thermo-tank, berth deck, frames #128-130, starboard.
10			Thermo-tanks, gun deck, frames #106-107, port, and #117-118, starboard.
11			General mess pantry, mess attendants wash room, sick bay and bath, operating room, dispensary and surgeons examining room.
12			Chief petty officers mess and wash rooms, laundry, crew's W.C. and wash room, blow-out and sanitation connections, and water heater in after battle dressing station.
13	Forward galley circuit.	Water heater drain to auxiliary exhaust.	Carbon dioxide system.
14	After water heater circuit.	Do.	Do.

tight compartments with pump rooms between, as shown in Plate I.

The engines are of the vertical, inverted-cylinder, direct-acting, four-cylinder, triple-expansion type, turning outboard when going ahead. The order of the cylinders, beginning forward, is forward low-pressure, high-pressure, intermediate-pressure and after low-pressure. All crank angles are 90 degrees, the sequence of cranks being high-pressure, intermediate-pressure, forward low-pressure, and after low-pressure.

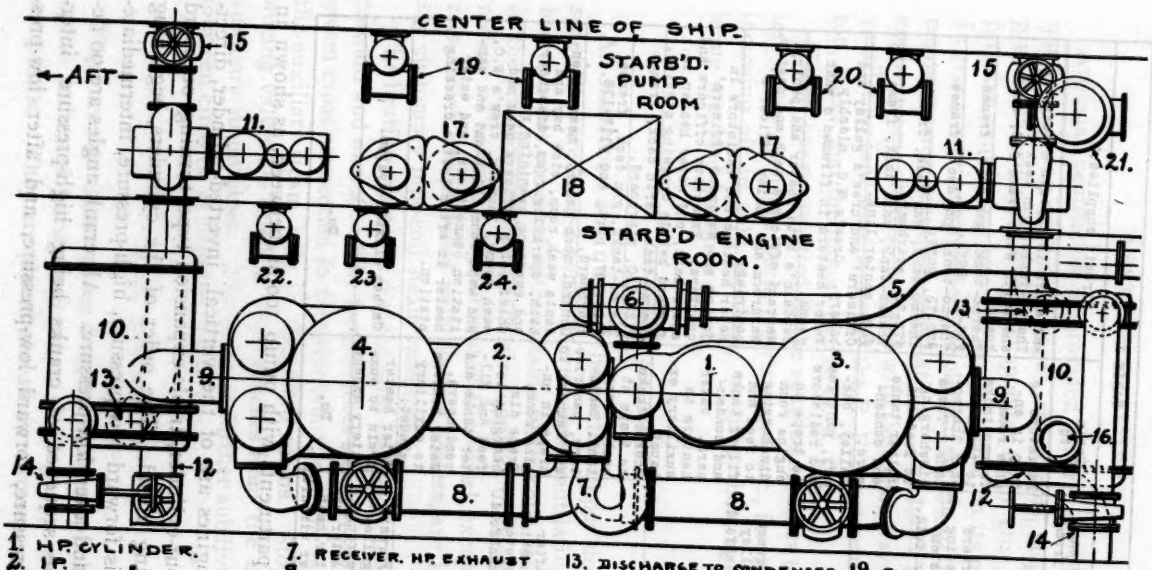

CENTER LINE OF SHIP.
←AFT—
STARB'D. PUMP ROOM
STARB'D ENGINE ROOM.
1. H.P. CYLINDER.
2. I.P. "
3. FORW'D L.P. CYLINDER.
4. AFTER L.P. "
5. 13" MAIN STEAM.
6. THROTTLE VALVES & EXP. JOINT
7. RECEIVER. H.P. EXHAUST
8. " I.P. EXHAUST
9. EXHAUST TO CONDENSERS
10. MAIN CONDENSERS
11. MAIN CIRC. PUMPS & ENG.
12. 21½" MAIN INJECTION
13. DISCHARGE TO CONDENSER
14. OVERBOARD DELIVERY
15. MAIN INJ. CROSS CONNECTION
16. DYNAMO EXHAUST; AFTER
17. MAIN AIR PUMPS
18. FEED & FILTER TANK
19. FIRE & BILGE PUMPS
20. MAIN FEED PUMPS
21. FEED WATER HEATER
22. SERVICE PUMP; OIL—
23. SERVICE & RETURN PUMP;
24. RETURN PUMP; OIL—

PLATE I.—MACHINERY DIAGRAM.

Bedplates.—The bedplates are of cast steel, in three sections each, bolted together and supported on keelson plates. Proper seatings and facings are provided for the main bearings, columns, etc.

Main Bearings.—The main bearings consist of a lower brass and cast-steel cap, each lined with white metal and cored for the circulation of cooling water.

Framing.—The engine frames are of the usual Navy type forged-steel columns, bolted to the bedplate and cylinders, and braced by suitable diagonal, cross and longitudinal stays.

Cylinders.—The cylinders and valve chests are of cast iron, fitted with working liners of close-grained cast iron as hard as can be properly worked. All cylinders, except the high-pressure, are steam jacketed around the working liners and at both ends.

Pistons.—All pistons are of conical design, those for the high-pressure cylinders being of cast iron, all others are of cast steel. The high-pressure followers are of cast iron, and those for the intermediate and low-pressure pistons are forged steel. The high and intermediate-pressure pistons have one solid packing ring each, those for the latter being cut at one point and bound solid with a tongue piece, and each low-pressure piston one ring, cut obliquely into three sections and fitted with tongue pieces at each cut, which permit slight movement. The intermediate and low-pressure packing rings are floated by springs. There are no springs for the high-pressure rings, which fit the pistons snugly. All packing rings are cast iron.

Piston Rods.—Each piston rod is tapered to fit its piston and secured by a locked nut. The lower end is fitted to a forged-steel crosshead, to which is bolted a cast-steel slipper, white-metal lined. The piston rods are of forged steel.

Crosshead Guides.—The go-ahead guides are of cast iron, hollowed for the circulation of cooling water. The backing guides are of cast steel and securely bolted to flanges on the go-ahead guides. The guides are supported by bolting to facings on the cylinders at the upper end, and to a cast-steel

girder of "I" section at the lower end; the girder being secured to the inboard engine columns.

Connecting Rods.—The connecting rods are of forged steel, forked at the top to span the crosshead and carrying the crosshead brasses, and "T" headed at the bottom to receive the crank-pin brasses.

Valve Gear.—The engines are equipped with the Stephenson double-bar link valve gear, fitted with Lovekin assistant cylinders. Piston valves are used throughout; there being one for each high-pressure and two for each intermediate and low-pressure cylinders.

Reversing Gear.—Each main engine is provided with a reversing engine of the vertical, direct-acting type, bolted to the high-pressure cylinder and connected through connecting rods to the reversing-shaft arms; the shaft, in turn, connecting to the main links by arms and suspension rods. Each reversing engine has a steam cylinder 16 inches in diameter by 22¾ inches stroke, and an 8-inch oil-controlling cylinder of same stroke, for taking up shock and for hand operation, a small hand pump being provided for the latter purpose. The gear is controlled by a floating lever operated at the working platform.

Turning Gear.—A 15-horsepower General Electric Co.'s motor is provided to operate the turning gear of each main engine. The motors are reversible and drive by worm gearing a second worm, which may be made at will to mesh with a worm wheel fitted on the crank shaft.

Each turning gear is also fitted for turning by hand.

Lubricating Gear.—All working and moving parts of the main engines, except the valve links and valve-stem guides, which are efficiently lubricated by combination sight and wick-feed oil-distributing boxes, located on the main cylinders, are lubricated by the forced-lubrication system, described elsewhere. The crosshead guides are provided with both gravity and forced lubrication.

Water Service.—Water service is provided for each main engine from the discharge pipe of the main circulating pump. This pipe has suitable branches to the various parts of the

main engine, thrust and spring bearings; the discharge being returned to the suction side of the same pump.

Working Platform.—The floors on the inboard side of each main engine, opposite the high and intermediate-pressure cylinders, are arranged to serve as working platforms.

Lifting Gear.—Efficient hand-operated lifting gear is provided for handling parts of the machinery for overhauling and repairing.

Main Engine Data.

Working pressure at H.P. chest, pounds gage	265
Revolutions per minute, designed	125
Indicated horsepower, total designed	24,800
Diameter of H. P. cylinder, inches	35
I.P. cylinder, inches	59
L.P. cylinders (2) inches	78
Stroke, inches	48
Ratio, I.P. to H.P.	2.84
L.P. to H.P.	9.93
L.P. to I.P.	3.50
Total expansions	12.63

	Per cent. of volume.		Linear, inches.	
Cylinder clearances:	Top.	Bottom.	Top.	Bottom.
Starboard, H.P.	7.226	9.586	$\frac{1}{4}$	$\frac{5}{16}$
I.P.	9.284	12.499	$\frac{3}{8}$	$\frac{1}{2}$
F.L.P.	9.990	12.572	$\frac{7}{16}$	$\frac{1}{2}$
A.L.P.	8.883	11.863	$\frac{7}{16}$	$\frac{15}{32}$
Port, H.P.	6.855	9.360	$\frac{9}{32}$	$\frac{5}{16}$
I.P.	9.507	12.261	$\frac{3}{8}$	$\frac{1}{2}$
F.L.P.	9.070	11.178	$\frac{7}{16}$	$\frac{7}{16}$
A.L.P.	9.171	11.438	$\frac{3}{8}$	$\frac{1}{2}$

Valves and valve settings:	H.P.		I.P.		L.P. (each).	
Number and type of valves	1 piston.		2 piston.		2 piston.	
Diameter of valves, inches	20½ top.		23 top.		30½ top.	
	$20\frac{3}{16}$ bottom.		$22\frac{15}{16}$ bottom.		$30\frac{7}{16}$ bottom.	
Travel of valves, inches	9		10		12	
Inside or outside steam	Inside.		Outside.		Outside.	
	Top.	Bot.	Top.	Bot.	Top.	Bot.
Width of port, inches	$3\frac{1}{2}$	$3\frac{1}{2}$	$3\frac{1}{2}$	$3\frac{1}{2}$	4	4
Steam opening, linear, inches	$2\frac{1}{16}$	$2\frac{1}{8}$	$2\frac{1}{8}$	3	$2\frac{1}{2}$	$2\frac{1}{16}$
Exhaust opening, linear, ins.	Full port.					
Steam lap, inches	$1\frac{13}{16}$	$1\frac{1}{4}$	$2\frac{1}{8}$	2	$3\frac{1}{8}$	$3\frac{1}{16}$
Exhaust lap, inch	$-\frac{7}{16}$	$+\frac{3}{32}$	$-\frac{9}{16}$	$+\frac{7}{16}$	$-\frac{13}{16}$	$+\frac{13}{16}$
Steam lead, linear, inch	$\frac{11}{32}$	$\frac{11}{32}$	$\frac{11}{16}$	$\frac{11}{16}$	$\frac{11}{16}$	1
Cutoff, decimal of stroke	83.4	77	80.3	72.3	68.5	58.5
mean	80.2		76.3		63.5	

Piston rods, diameter, outside, inches	07½
axial hole, inches	1¼ H.P. and I.P., 3¼ L.P.
Connecting rods, diameter, crosshead end, inches	07½
crank end, inches	09
axial hole, inches	1¼ H.P. and I.P., 3¼ L.P.
length between centers, inches	96
crank ratio	4
Diameter of throttle valve, inches	13
1st receiver pipes (2), inches	14
2d receiver pipe (2), inches	23
main exhaust pipes (2), inches	30½
live-steam pipe to I.P. receiver, inches	02½

SHAFTING AND BEARINGS.

There are two lines of shafting, each consisting of a crank shaft in two sections, a thrust shaft, two pieces of line shafting, one forward and the other astern of the thrust shaft, a stern-tube shaft and a propeller shaft, all supported by suitable bearings.

Crank Shafts.—The crank shafts are solid-forged hollow shafts, in two sections each, bolted together by disc couplings, and carried in seven main bearings. The forward section includes the H.P. and F.L.P. cranks, which are opposite, as are also the I.P. and A.L.P., on the after section. All crank angles are ninety degrees, the two sections being connected with the H.P. crank leading, followed by the I.P. crank.

Thrust Shafts.—The thrust shafts are fitted to the usual thrust bearings of the adjustable horse-shoe type, with steady bearings at each end for supporting the shafts. The shafts are hollow with thrust collars and coupling discs forged integral with the shafts.

Line Shafts.—These shafts are hollow forged, with disc couplings at either end, and each is carried by one steady bearing.

Stern-Tube Shafts.—The stern-tube shafts are covered with a composition casing within the stern tubes and at bearings. They are hollow forged, secured to the line shafts at inboard end by special disc couplings, and to the propeller shafts at outboard end by sleeve couplings. Each stern tube is provided with three lignum vitae lined bearings.

Propeller Shafts.—Each propeller shaft is carried by two lignum vitae lined bearings, fitted to the forward and after struts. The shafts are hollow forged, taper turned at after end to suit the propeller hub, and composition bushed at the bearings.

Inboard Coupling.—The inboard coupling consists of a collar secured to the stern-tube shaft by four keys. Two ring segments are inserted through a cross slot in the forward end of the collar and fit a groove turned in the end of the shaft, preventing its backing out. Collar and segments are through bolted to the line-shaft coupling disc.

Outboard Coupling.—The outboard coupling is of the solid-sleeve type, taper bored for reception of shafts, and secured to each shaft by two feather keys and one cross key.

Shaft Data.

Crank shafts, length, forward section, feet and inches	16–03½
after section, feet and inches	18–02
diameter, inches	18½
axial hole, forward section, inches	10
after section, inches	08
number of cranks	4
throw of cranks, inches	24
crank angles, degrees	90
Crank pins, length, inches	21
diameter, inches	19½
axial hole, inches,	F.L.P. 10⅜, H.P. 11½, I.P. 8, A.L.P. 8
webs, width, inches	21
thickness, inches	F.L.P. 9⅜, H.P. 10⅜, I.P. 11, A.L.P. 11⅝
Forward line shafts, length, feet and inches	14–07½
diameter, inches	17½
axial hole, inches	08⅝
Thrust shafts, length, feet and inches	15–05
diameter, inches	17½
at bearings, inches	17⅝
axial hole, inches	08⅝
collars, number	14
thickness, inches	02
space between, inches	04
outside diameter, inches	27½
inside diameter, inches	17⅝
bearing surface, square inches, ahead	4,502.68
astern	4,181.06

After line shafts, length, feet and inches	17–04⅜
diameter, inches	17½
axial hole, inches	08¼
Stern-tube shafts, length, feet and inches	60–05⅛
diameter, inches	17½
axial hole, inches	08¼
Propeller shafts, length, feet and inches	54–04⅞
diameter, inches	18
axial hole, inches	09
Coupling discs, diameter, inches	33
thickness, inches	04
Inboard coupling, diameter of collar outside, inches	33
inside, inches	17½
length of collar, inches	16½
thickness of segments (inshear), inches	01½
Coupling bolts, number each coupling	8
diameter (taper) at face of coupling, inches	03⅛
Outboard couplings, length of sleeve, inches	69
diameter of sleeve, inches	24

Bearing Data.

Main bearings (white-metal lined):	
Number each engine	7
Diameter, inches	18½
Length, inches	two of 13½, four of 21½, and one of 26½
Thrust bearings (white-metal lined bearings and shoes):	
Steady bearings, number each	2
diameter, inches	17½
length, inches	18
Thrust shoes, number each	14
effective surface, square inches, ahead	2,660
astern	2,470
Line-shaft bearings (white-metal lined):	
Number each engine	2
Diameter, inches	17½
Length, inches	24
Stern-tube bearings (lignum vitae lined):	
Number each engine	1
Forward bearing, diameter, inches	19½
length, inches	63
Middle bearing, diameter, inches	19½
length, inches	30
After bearings, diameter, inches	19½
length, inches	60
Strut bearings (lignum vitae lined):	
Number each engine	1
Forward bearing, diameter, inches	20
length, inches	36
After bearing, diameter, inches	20½
length, inches	61

PROPELLERS.

There are two three-bladed propellers of the adjustable-pitch, detached-blade type. They are of manganese-bronze and the blades are secured to the hubs by nine $3\frac{1}{2}$ tap bolts each. The blades are machined true to pitch and the hubs have a taper fit on the shafts, and are secured by a key and nut.

Propeller Data.

Diameter of propeller, feet and inches	18–03
hub, feet and inches	4–08
Pitch as set, feet and inches	starboard 19–5¼, port 19–05 $\frac{5}{32}$
adjustable from, feet and inches	18–5¼ to 20–05¼
Ratio of diameter to pitch	starboard 1.065, port 1.0604
Area, projected, square feet	77.85
helicoidal, square feet	91.65
disc, square feet	261.58
Ratio, projected to disc area	0.2976
helicoidal to disc area	0.3504
Heighth of lower tip of blade above keel, inches	06 $\frac{5}{16}$
Immersion of upper tip of blade, feet and inches	10–06 $\frac{9}{16}$

MAIN CONDENSING APPARATUS.

Main Condensers.—There are four main condensers, one for each low-pressure cylinder, with tubes rolled into the tube sheet at one end and gland packed at the other. The two forward condensers also serve the after dynamo plant, which has no independent condenser. The principal dimensions follow:

	Forward condensers.	After condensers.
Inside diameter, feet and inches	8–02	7–08
Thickness of shell (steel), inch	00½	00½
Length between tube sheets, feet and inches	7–06	7–06
Thickness of tube sheets, inches	01½	01½
Tubes number	5,810	5,060
diameter, outside, inch	00¾	00¾
thickness, inch	0.065	0.065
Cooling surface, square feet	7,128	6,208
Diameter of main exhaust nozzles, inches	30½	30½
dynamo exhaust nozzle, inches	20	.....
air-pump suction, wet, inches	07	07
dry, inches	07½	07½
auxiliary exhaust nozzle, inches	10	.....
circulating-water inlet and outlet, inches	21½	21½

Pumps.—The main air and circulating pumps and their connections are described in Table II.

Feed and Filter Tank.—A feed and filter tank of 4,700 gallons capacity is located in each pump room. The filter chamber is in the top of the tank and has a capacity of about 800 gallons. The filter has an inner bottom of loose perforated plates and is divided into three compartments, in which is placed the filtering material, by vertical division plates. These partitions are so arranged that the water in passing through the filter will flow under and over in succession, thus assuring the filtering material being always submerged.

Each tank is provided with the following connections:

2 8½-inch main air-pump discharges;
1 21-inch stand pipe;
1 11½-inch cross connection between stand pipes;
1 10-inch auxiliary-feed suction main from cross connection;
2 5½-inch main-feed suctions from cross connection;
1 8-inch overflow;
3- and 2-inch vapor pipes (combined into one 3-inch pipe).

PUMP-ROOM AUXILIARIES.

Pumps.—A description of all pumps and their connections is given in Table II.

Feed-Water Heaters.—Two Schutte-Koerting, spirally-corrugated, film-feed water heaters are provided—one in each pump room. They have 277.38 square feet of heating surface each, and are connected to the main feed discharge lines only. The heating agent is the exhaust steam, a back pressure being kept in the auxiliary exhaust line for this purpose by means of a spring-relief valve at each connection to the forward main condensers, opening toward the condenser.

FORCED-LUBRICATION SYSTEM.

All working and moving parts of the main engines, except the valve links and valve-stem guides, are fitted with forced lubrication.

TABLE II – PUMPS AND CONNECTIONS – U.S.S. OKLAHOMA

No.	Pumps	Size (Ins.)	Type	Suction Pipes from			Discharge Pipes to			Location
				Ins.			Ins.			
4	Main Air.	2(12x26x18 21)	Twin Vertical	10[a]	Condenser.		8½	Feed Tank.		Two in each Pump Room.
4	Main Circulating.	36,000 Gals. per min. at 600 R.P.M. 24 Stroke	Hor. Centrifugal, Vert., Compound, Eng. 24x13x10".	21½	Sea.		21½	Condenser.		Do.
				15½	Drainage.		5	Oil Cooler and Water Service.		
				21½	Cross-Connection.					
4	Main Feed.	15x10x24	Vert., Piston, Double-Acting, Single, "Warren."	5½	Feed Tank Standpipe Cross-Connection.		5	Main Feed Discharge.		Do.
1				2½	Reserve Feed Tanks.					
3	Aux. Feed.	Do.	Do.	5½	Feed Tank Cross-Conn.		5	Main Feed Main.		One in each Boiler Room.
				2½	Reserve Feed Tanks.		2½	Hose Connection.		
				2½	Hose Connection.		2½	Reserve Feed Tanks.[1]		
							8½	Each Boiler in Compartment in which Pump is located.		
4	Main Fire and Bilge.	12x10x18	Do.	5	Sea.		5	Fire Main.		Two in each Pump Room.
				5	C. & R. Drainage.		5	Overboard.		
				1½	Hose Connection.		5	Sanitary.[2]		
				4	Hose Connection.[2]		2½	Hose Connection.		
							4	Hose Connection.		
3	Fireroom Fire and Bilge.	Do.	Do.	5	Sea.		5	Fire Main.		One in each Boiler Room.
				5	C. & R. Drainage.		5	Overboard.		
				1½	Hose Connection.		5	Sanitary.		
				4	Flange Connection.[3]		2½	Hose Connection.		
							4	Flange Connection.[3]		
1	Dynamo Cond. Circulating.		Jeanesville Double-Inlet, Volute; Terry Turbine.	13	Sea.		13	Dynamo Condenser.		Dynamo Cond. Room.
1	Dynamo Cond. Air.	10x20x12	Vert. Twin-Cylinder, Beam.	7	Dynamo Condenser.		6	Dynamo-Room Hot Well.		Do.
1	Dynamo-Room Hot Well.	4½x3x6	Vert., Piston, Double-Acting Single "Warren."	2½	Dynamo-Room Hot Well Tank.		2	Aux. Feed-Suction Main.		Do.
2	Evapora-tor Feed.	5½x6x6	Vert., Piston, Light-Service, Double-Acting, Single, "Warren."	2½	Sea.		2	Evaporators through Feed Heaters.		Evaporator Room.
				2½	Overboard Disch. from Distillers					
2	Brine from L.P. Evap.	Do.	Do.	1½	L.P. Evaporators.		2	Overboard.		Do.
				1½	Sea.					
2	Distiller Circulating		Worthington Double-Inlet, Hor., Volute; Terry Turb.	6	Sea.		5	Distillers.		Do.
2	Distiller Fresh Water.	4½x5x6	Vert., Piston, Light-Service, Double-Acting, Single, "Warren."	2½	Bottom of Distillers and Test Tank.		2½	Fresh Water Tanks, Main Feed Tanks and Reserve Feed Tanks.		Do.
2	Forced Lubrication Service.	6x7x8	Vert., Piston, Double-Acting, Single, "Warren."	3½	Supply Tanks.		2½	Main Engines.		One in each Engine Room.
							1½	Thrust Bearings.		
							1½	Circ. Pump Engines.		
							½	Gravity & Supply Tanks.		
2	Forced Lub. Service and Return.	Do.	Do.	3½	Supply Tanks.		2½	Main Engines.		Do.
				3½	Oil Well.		3	Settling Tank and Storage Tanks.		
				2	Bottom of Oil Well, (Independent Suction)		1	Overboard.		
							1½	Thrust Bearings.		
							1½	Circ. Pump Engines.		
							½	Gravity & Supply Tanks.		
2	Forced Lubrication Return.	7½x8½x8	Do.	4	Oil Well.		3½	Settling Tank and Storage Tanks.		Do.
				3	Storage Tanks.					
6	Fuel-Oil Service.	5½x4x8	Vert., Piston, Double-Acting, Duplex, "Warren."	3	Fuel-Oil Main.		1½	Burner Supply.		Two in each Boiler Room.
4	Fuel-Oil Booster.	6x8x8	Vert., Piston, Double-Acting, Single, "Warren."	5	Tank Manifold.		5	Fuel-Oil Main.		Two in each End Boiler Room.
							5	Tank Manifold.		
							6	Overboard.		
1	Brine.	4½x5x6	Vert., Piston, Double-Acting, Single, "Blake."	2	Brine Tank.		2	Scuttle Butts.		Refrigerating Room.
2	Sanitary		Centrifugal, Motor Driven, Horizontal.	6	Sea		6	Sanitary System.		Upper Platform Fresh Pump Room, Aft.
2	Fresh Water.		Do.	3½	Fresh Water Tank.		3	Fresh Water System.		Hold Pump Room, Forward.
1	Secondary Drainage		Do.	7	Sea.		5	Drainage.		Do.
							5	Overboard.		
3	Main Drain.		Centrifugal, Motor Driven, Vertical.	13½	Main Drain.		12	Overboard.		One in each Boiler Room.

[1] No. 2 Boiler Room only. [2] One Pump in each Pump Room. [3] No. 1 Boiler Room only.
[a] 7" Wet and 7½" Dry Suctions from Condensers Combine into One 10" Pipe.

The installation is divided into two complete and independent systems, starboard and port, cross connections being provided, however, between the systems for emergencies. Each system comprises three pumps (see Table II), two 500-gallon oil-storage tanks, one 500-gallon oil-supply tank and one 500-gallon oil-settling tank, together with the necessary piping and fittings.

The crank pits are of oiltight construction, fitted with an oil-drain well, and forming an oil trough at the base of each engine to catch all the oil. The entire engine is incased with a light galvanized sheet-steel casing, to prevent splashing and waste of oil, which is carried up to within about eighteen inches of the bottoms of the cylinders.

The plant functions as follows: one pump draws the oil from the supply tank and discharges same to the main engine, thrust-bearing and main circulating-pump engines on its side of the ship. The oil enters the main engine circuits at the main bearings through holes in the caps. An annular groove in the center of each main bearing provides for the proper distribution of the oil, part of which lubricates the bearings, the balance passing through a radial hole in each journal, in wake of the groove to the crank-shaft axial holes, the openings at ends of each axial hole being closed by oiltight cover plates. From the axial holes the oil is forced through radial holes to the eccentric straps and crank-pin axial holes, and through similar radial holes to the surface of the crank pins. The crank-pin bearings are grooved like the main bearings, all oil not used for crank-pin lubrication passing up through the connecting rods, to the crosshead bearings and guides. After performing its function the oil escapes at the ends of all bearings and drains to the crank pit, where it collects in the drain well and is pumped by a second pump, through oil filters, to the main supply tank, thus completing its cycle, which is indefinitely repeated as described. The third pump performs auxiliary duties.

Pressure gages and thermometers are provided at each main

bearing. The system is generally operated at about 50 pounds pressure.

BOILERS.

There are twelve Babcock & Wilcox water-tube boilers, arranged in three separate watertight compartments, as shown in Plate II.

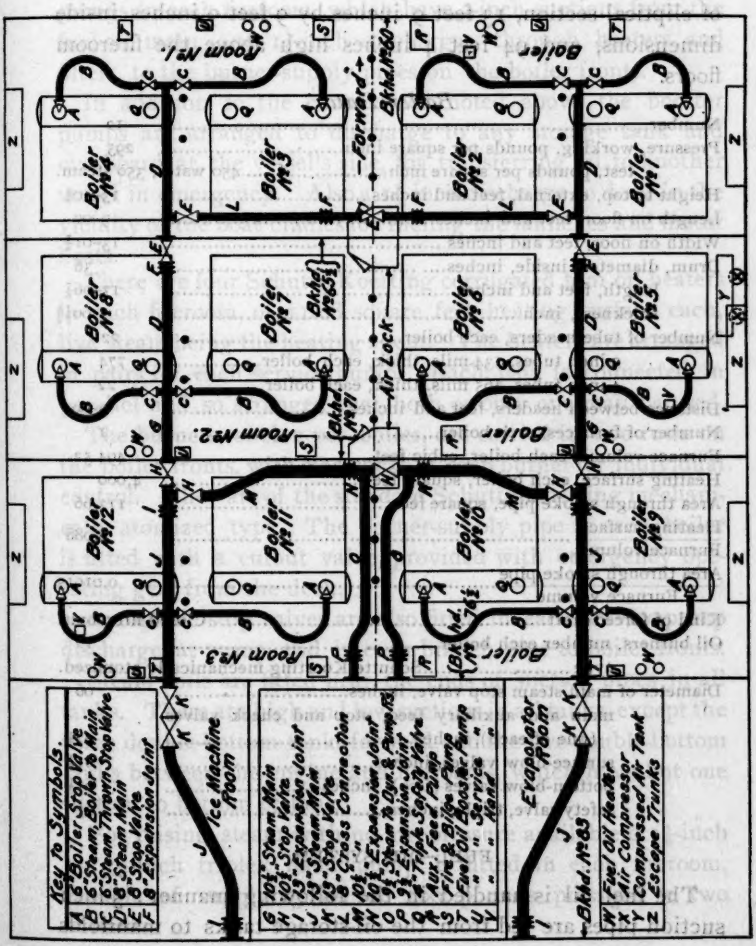


PLATE II.

The boilers have a total heating surface of 48,000 square feet and are designed to operate the entire machinery plant at full power, with an average air pressure in the ash pits of not more than six inches of water. They are equipped for burning fuel oil.

Uptakes of the usual design are provided, all uniting into a common smoke pipe at the main deck. The smoke pipe is of eliptical section, 13 feet 9 inches by 9 feet 9 inches inside dimensions, and 94 feet 4 inches high above the fireroom floors.

Boiler Data.

Number	12
Pressure, working, pounds per square inch	295
test, pounds per square inch	450 water, 350 steam.
Height to top, external, feet and inches	13–10⅛
Length on floor, feet and inches	9–00
Width on floor, feet and inches	13–01½
Drum, diameter, inside, inches	36
length, feet and inches	13–06½
thickness, inch	00$\frac{29}{32}$
Number of tube headers, each boiler	22
2-inch tubes, 134 mils. thick, each boiler	774
4-inch tubes, 165 mils. thick, each boiler	22
Distance between headers, feet and inches	9–00
Number of furnaces, each boiler	1
Furnace volume, each boiler, cubic feet	494.52
Heating surface, each boiler, square feet	4,000
Area through smoke pipe, square feet	113.66
Heating surface / Furnace volume	8.088
Area through smoke pipe / Furnace volume	0.01915
Kind of forced draft	Closed firerooms.
Oil burners, number each boiler	12
type	Schutte-Koerting mechanically atomized.
Diameter of main steam stop valve, inches	06
main and auxiliary feed, stop and check valves (one of each), inches	02½
surface-blow valves, inches	01½
bottom-blow valves (two), inches	01½
safety valve, triple, inches	04

FUEL–OIL SYSTEM.

The fuel oil is handled in the following manner: 5-inch suction pipes are led from the oil-storage tanks to manifolds

in firerooms Nos. 1 and 3. In each of these firerooms are located two light-service booster pumps, having 5-inch suction connections to the manifolds, and 5-inch discharge, through strainers, to the fuel-oil main, which is 5 inches in diameter, and extends through the firerooms, in continuous loop, the necessary cutout valves being fitted for isolating any portion of the main not in use. There are two heavy-pressure duplex pumps in each fireroom, which have 3-inch suctions from the fuel-oil main, and 1½-inch discharges, through heaters and filters, to the burner-supply pipes on the boiler fronts.

In addition to the connections noted above the booster pumps are arranged to discharge to any storage tank and overboard at the vessel's side, for transferring oil to another vessel in emergency. Also a 1½-inch discharge to deck in the vicinity of the boat cradles for fueling the launches and motor boats.

There are four Schutte-Koerting corrugated film oil heaters in each fireroom, of 11.08 square feet heating surface each, live steam being the heating agent. The heaters are arranged in pairs for each service pump. Each pair is connected in parallel and so arranged that both or only one may be used.

The burners, twelve per boiler, are carried in two rows on the boiler fronts, with stop cocks at each burner for individual control. They are of the standard Schutte-Koerting mechanically atomized type. The burner-supply pipe to each boiler is fitted with a cutout valve, provided with emergency operating gear from the deck.

Automatic stop valves are also fitted in each service-pump discharge, at pump, and in each branch pipe to boiler fronts.

Steam coils are fitted near the ends of suction pipes in all tanks. There are high and low suctions in all tanks, except the three double-bottom tanks forward and the two double-bottom tanks between the engine and firerooms, which have but one suction to bottom of tank.

For raising steam with no air pressure available a 1¼-inch by 3½-inch triplex hand pump is fitted in each fireroom, capable of supplying oil at 200 pounds pressure to two burners.

The characteristics of the fuel-oil pumps and connections are given in Table II.

PNEUMERCATOR SYSTEM.

For ascertaining at any time the amount of fuel oil in the various double-bottom tanks, the Parks pneumercator system is installed.

The installation consists of a small semi-spherical balance chamber placed at a predetermined location near the bottom of each tank. The interior of the balance chamber is in communication with the tank, through a hole in the side of the chamber, and a small pipe connects the top of the chamber with its recording instrument in the engine rooms and fire-rooms at the fuel-oil manifolds. The recording device consists of a glass mercurial tube provided with scale, calibrated to suit the tank to which it is connected.

The instrument operates on the following principle. The pressure due to the head of oil in the tank compresses the air in the chamber and connecting pipe line, which in turn causes the mercury to rise or fall in the tube in direct ratio to the pressure exerted. The tank contents is read off on the scale, as in the case of an ordinary thermometer.

In order to guard against overflowing the tanks when taking aboard fuel oil, an annunciator is installed in connection with the recording device for indicating and signalling when any tank is 95 per cent. full.

FIREROOM AUXILIARIES.

Forced-Draft Blowers.—Each fireroom has four forced-draft blowers. They are located in specially constructed blower rooms just below the protective deck and above the working flat in front of the boilers.

The fans are 29¼ inches outside diameter, Sturtevant multivane, double-inlet type, each capable of delivering 15,000 cubic feet of air per minute against a static pressure of 6 inches of water, when running at 970 revolutions per minute.

The blowers are driven by direct-connected General Electric Company's motors of 40 H.P. each, controlled from the fireroom working level and the blower room at will. Air is supplied from the fireroom ventilators, which are closed at the bottom when under forced draft.

Fireroom Hoist.—The port ventilator in the middle fireroom is fitted with a hoisting engine, the necessary guides, wire ropes, sheaves, etc., being installed for the purpose. The engine is operated from the upper deck. It is a 4½-inch × 4½-inch, two-cylinder, reversible engine, made by the Hyde Windlass Co., and it is located in the fireroom hatch at the berth deck.

Pumps.—The auxiliary feed pumps and fire and bilge pumps and their connections are described in Table II.

MAIN STEAM PIPING.

The main steam piping is arranged in two symmetrical systems, one on each side of the vessel. The two lines are cross-connected in the forward and after firerooms by 8- and 10½-inch connections, respectively. The branches from the boilers are 6 inches in diameter each, and the lines proper are 8 inches in the forward fireroom, increasing to 10½ and 13 inches at each successive boiler connection, the latter size continuing to the main engine's throttle valves. The piping is seamless-drawn steel.

The arrangement of the main steam piping is shown in Plates I and II.

AUXILIARY STEAM PIPING.

The auxiliary steam pipe is seamless-drawn steel.

From the main steam cross connection in the after fireroom there are two 9½-inch connections leading aft through the after dynamo room and pump rooms, with branches to dynamo turbines and auxiliary machinery. The pipe is reduced to 7½-inch diameter in the pump rooms, where it passes aft and through the centerline bulkhead forming a connecting loop

between the two lines. Stop valves are fitted at all bulkheads and at branch connections to auxiliaries.

In the firerooms the auxiliary steam piping consists of branches from the main steam piping with sub-branches to the various auxiliaries.

The forward dynamos and windlass take their steam off the main steam cross-connection in the forward fireroom.

Stop valves are fitted in all branches and sub-branches as required.

AUXILIARY EXHAUST PIPING.

An 8-inch copper auxiliary exhaust pipe is led throughout the machinery spaces with branches elsewhere as required for the various auxiliaries. Connections are provided to direct the exhaust steam into either forward main condenser, either feed-water heater, or into the atmosphere through the after escape pipe at will. There are also connections for admitting the exhaust steam to the L.P. receivers.

Stop valves are fitted in all branches at the main.

ESCAPE PIPES.

From each safety valve a 6-inch pipe is led to an 11-inch fore-and-aft line, on its side of the ship, between the firerooms. The two 11-inch lines have 14-inch vertical branches in middle fireroom, leading through the deck and connecting to a 19-inch escape pipe on the after side of the smoke pipe.

In the middle fireroom there are two 8-inch branches, port and starboard, from the auxiliary exhaust to the escape pipe for atmospheric exhaust. A stop valve is fitted at each connection to the escape pipe.

There is also a 10-inch atmospheric exhaust connection, fitted with automatic valve, from each dynamo turbine to the escapes pipes.

All the above piping is copper.

MAIN AND AUXILIARY FEED SYSTEMS.

Each main feed pump has a 5½-inch independent suction from the main feed tank cross-connecting pipe. The pumps discharge *via* the feed-water heaters, or by-pass same if desired, to the boilers. The combined discharge from each pair of pumps is 6 inches, uniting in the after fireroom into an 8½-inch connection leading forward and diminishing in size as it advances to 5 inches in the forward fireroom.

The auxilary-feed suction main is taken off the 11½-inch feed tank's cross-connecting pipe. This main is 10 inches in diameter, reducing in size to 5½ inches, as it leads forward to the auxilary feed pumps, each pump having a 5½-inch suction connection. The auxiliary feed pumps discharge direct to the boilers in their respective compartments, or into the main feed line to any boiler. The discharge connections are 5 inches in diameter at the pumps.

All branch, main and auxiliary feed pipes to the boilers are 2½ inches in diameter.

The feed piping is seamless-drawn copper.

INTERIOR COMMUNICATION.

The customary engine and fireroom telegraphs, gongs, telephones, voice tubes, etc., are fitted for transmitting orders and signaling to the various machinery compartments and other parts of the vessel.

AIR-COMPRESSOR PLANTS.

There are two Ingersoll-Rand turbo-air compressors placed in the compartment abreast of the forward dynamo room, starboard side. Each compressor has a capacity of 2,350 cubic feet of free air per minute at 100 pounds pressure when running at 6,000 r.p.m.

In the middle fireroom are two 11-inch by 11-inch by 12-inch water-cooled Westinghouse steam-driven reciprocating air compressors and air reservoir of 27,500 cubic inches capa-

city, for use in running pneumatic tools in the engineering department and blowing soot off the boiler tubes. Each compressor has a capacity of about 45 cubic feet of free air per minute at 150 pounds pressure.

A pneumatic main is led throughout the machinery space, with branches to the general workshop, evaporator and dynamo rooms, from which the connections for pneumatic tools and blowing soot off boiler tubes are taken.

EVAPORATING AND DISTILLING APPARATUS.

The evaporator room is on the lower platform between the engine and firerooms, starboard side. There are four evaporators, two distillers, two feed-water heaters, and two, each, distiller circulating, evaporator feed, brine and fresh-water pumps, together with their accessories. The plant has a combined capacity of 25,000 gallons of water per 24 hours, and is arranged to operate in double effect. The pumps and their connections are given in Table II.

Evaporator Data (each).

Type	Schutte-Koerting, vertical U-tubes.
Diameter, inside, inches	38½
Height over all, feet and inches	7–05½
Tubes, number	54
diameter of pipe, inches	01¼
thickness of pipe, inch	0.083
Heating surface, square feet	117.78
Diameter of steam connection, inches	02
vapor nozzle, inches	07
feed valve, inch	02
blow valve, inches	02

Distiller Data (each).

Type	Horizontal, curved tube.
Diameter, inside, inches	20½
Length, over all, feet and inches	5–07½
Tubes, number	289
diameter, inch	00⅝
thickness, inch	00.049
Cooling surface, square feet	200
Diameter of circulating water inlet and outlet, inches	05
vapor inlet, inches	07
drain, inches	03

Evaporator Feed-Water Heater Data (each).

Type	Bureau, U-tube.
Tubes, number	12
diameter, inside, inches	02½
thickness, inch	0.083
Heating surface, square feet	26.21
Diameter of feed inlet and outlet, inches	02
vapor inlet and outlet, inches	07

GENERAL WORKSHOP.

The following machine tools, each driven by its own electric motor, are installed in the machine shop on the berth deck, between the engine-room hatches.

No.	Description.	Make and H.P. of motor.
1	28-inch by 48-inch swing, extension-gap lathe; E. Harrington and Sons Co.,	Reliance, 7½ H.P.
1	14-inch swing lathe; Prentice,	Reliance, 2 H.P.
1	14-inch swing lathe; Prentice,	Reliance, 2 H.P.
1	15-inch column shaper; Potter and Johnson,	Reliance, 2 H.P.
1	30-inch radial drill; Rand,	Reliance, 2 H.P.
1	16-inch sensitive drill; Willey,	Reliance, ½ H.P.
1	Universal milling machine; No. 1, Kempsmith,	Reliance, 2 H.P.
1	Floor grinder; Willey,	Reliance, 2 H.P.
1	Portable cylinder-boring machine; H. B. Underwood and Co.,	Reliance, 2 H.P.

BLACKSMITH SHOP.

A blacksmith shop is provided on the berth deck, in the forward boiler hatch. It is equipped with one portable and one permanent forge, together with anvil and all necessary tools and fittings. The permanent forge is fitted with an electrically-driven blast fan.

FOUNDRY.

A small foundry is installed on the berth deck, inside the after boiler hatch. Its outfit consists of a small oil-burning crucible furnace together with adequate allowance of crucibles and the necessary apparatus for handling and pouring the metal.

PIPE AND COPPERSMITH SHOP.

The pipe and coppersmiths' equipment is located in the foundry space and consists of the following:

1 hand pipe-cutting and threading machine;
1 tinner's combined punch and shears;
1 tinner's slip roll;
1 beading machine;
1 tinner's cornice brake;
vises, work-benches, etc.

ELECTRIC PLANT.

There are two dynamo rooms, one just forward of the forward boiler room and the other just aft of the after boiler room. They are on the lower platform level.

The distribution rooms, two in number, are located on the upper platform, one over each dynamo room. They contain the lighting, power and searchlight distribution boards only. The generator boards are in the dynamo rooms.

The generator installation consists of four 6-pole, compound-wound, 300-kilowatt, General Electric generators, two in each dynamo room, each driven by a two-stage horizontal Curtis turbine. Each generator will deliver at normal load 2,400 ampères of current at 125 volts, when running at 1,500 revolutions per minute. The generators are capable of delivering one-third overload for two hours without injury.

Steam for the after generators is taken off the auxiliary steam pipe in the after dynamo room, and that for the forward generators from the main steam cross connection in the forward fireroom. The branch to each generator is 4 inches and fitted with stop, reducing and throttle valves.

The generators are arranged with condensing and atmospheric exhaust, the latter connections, *via* escape pipe, are 10 inches and fitted with automatic valves permitting instantaneous change over in emergency.

The after pair of generators exhaust into the forward main condensers, each connection being 20 inches, and the forward generators have an independent condensing apparatus, located below the generators in a separate room in the hold, with access hatch between the two levels. The condensing apparatus consists of one condenser and vacuum augmenter (data below), one, each, independent air pump, circulating pump, hotwell-tank pump and hotwell tank. The pump data is given in Table II.

The dynamo condenser also handles the exhaust from the turbo-air compressors.

Dynamo Condenser Data.

Number	1
Inside diameter, feet and inches	5–05
Thickness of shell (steel), inch	00 5/16
Length between tube sheets, feet and inches	7–01
Thickness of tube sheets, inch	01
Tubes, number	2,615
diameter, outside, inch	00 5/8
thickness inch	00.065
Cooling surface, square feet	3,066
Diameter of exhaust nozzles, generators (2), inches	20
air compressors, inches	18
air-pump suctions, inches	07
circulating-water inlet and outlet, inches	13

Augmenter Condenser Data.

Number	1
Inside diameter, inches	16
Thickness of shell (composition), inch	00 5/16
Length between tube sheets, inches	39
Thickness of tube sheets, inch	00 3/4
Tubes, number	174
diameter, outside, inch	00 5/8
thickness, inch	00.049
Cooling surface, square feet	92.5
Diameter vapor inlet, inches	04 1/2
air-pump suction, inches	05
circulating-water inlet and outlet, inches	03

TRIALS.

The trials were run January 16 to 28, 1916, inclusive.

The official speed and revolution curve gave the following mean revolutions per minute of the propellers as necessary to attain the speeds desired for the various trials:

Speed in knots.	R.p.m.
20.5,	123.1
19,	113.8
15,	87.6
10,	59.65

TESTING OF METALS AT THE ENGINEERING EXPERIMENT STATION.

By D. J. McAdam, Jr., Associate.

In the testing of metals at the Naval Engineering Experiment Station conclusions are based, whenever possible, on the combined results of three general methods of investigation, namely, chemical analysis, physical tests and metallographic examination.

If the results of the two latter methods are to be of value the chemical composition of the metal must be known. In the present article, however, the methods of obtaining the chemical composition will not be considered, and attention will be confined to the subject of metallographic examination and physical testing.

METALLOGRAPHY.

The metallographic method of investigation, especially the use of the microscope in the study of metals, is of comparatively recent application. Though the first use of the microscope in investigation of metals by Sorby was made in 1863, it was not until 1885 that he discovered the nature of the constituent of steel known as pearlite, and it was not until after the beginning of the present century that the microscopic examination of metals became of general interest. In the last fifteen years this method has undergone a remarkable development, until today it is in use in nearly every works or laboratory where metals are manufactured or tested.

As in all new branches of science, there are varying views of the nature and scope of metallography. There is a prevalent opinion that it is merely microscopy as applied to metals. According to this view, anyone who can polish and etch metals

and recognize a few types of microscopic structure is a metallographist. This view of the subject, however, is entirely too narrow. Metallography is not limited to the use of the microscope. It makes use of any physical phenomenon that may throw light on the structure of a metal and on the relations of this structure to the processes of manufacture. For this purpose, in addition to heating and cooling curves and other temperature measurements, determinations of electrical conductivity, potential, volume, hardness, etc., have proved of great value.

Metallography, therefore, is a science. One authority defines it as "Physical Metallurgy;" another gives it the even broader definition of "Science of Metals." It is a branch of physical chemistry, yet in its practical application it is a branch of metallurgy. It deals with the microscopic structure of metals in relation to the laws of crystal origin, growth and transformation. As the geologist, from observation of regions on the earth's surface, can read traces of its history, so the metallographist, examining a small piece of metal, can read traces of its processes of manufacture even as far back as the molten condition. To do this, however, he must bring to his aid chemistry, physics and metallurgy, even as the geologist uses chemistry, physics and biology.

Metallography has solved many problems in regard to the ultimate structure of metals. The solution of these problems has advanced the subject to the point where it is of great use both to the manufacturer and to the user of metals. The manufacturer uses, or should use, metallography to assist him in producing good material. The buyer uses metallography to assist him in determining whether or not any particular metal is suitable for the purpose intended. Moreover, in addition to its use in the manufacturing and testing of metals, wider fields of investigation are being continually opened up by the progress of the science.

At the Experiment Station, in addition to the great number of routine tests of metals, an attempt is being made to investigate some of the many problems in regard to the effects

of heat treatment on microscopic structure and physical properties. Before discussing the subject of research, however, a description will be given of the method and apparatus used at the Station.

Fig. 1 is a general view of the metallographic laboratory. In this room, 17 feet × 19 feet in size, are the polishing machines, furnaces for heat treatment, microscope, camera, work bench, tables for etching reagents, typewriters and multigraph, desks, shelves for storage of specimens, and much other apparatus of various kinds. Here all the purely metallographic work is done, including the preparation of reports. There is also a dark room for the development of negatives and prints.

Routine tests of metals at this Station may be divided into two general classes: (*a*) Tests of material before acceptance, that is, for inspection purposes; (*b*) Tests of material that has failed in service.

INSPECTION TESTS.

Recent Navy Department specifications, 49S2*b*, require metallographic examination of all forgings used for shafting and couplings for naval vessels. In many other cases, also, metallographic examination is required by the terms of the contract.

For this examination specimens, in number depending on the size of the forging, are sent from the various inspection offices to this Station for examination. The number and location of these metallographic specimens are prescribed in specifications 49S2*b*. In addition to ends of broken tensile specimens other specimens, taken from various parts of the forging, are usually required. An examination is made at once and a preliminary report is sent out, usually within 24 hours of the time of receipt of the specimens. A supplementary report, including photomicrographs, is submitted later. If the material does not have the required uniformity and fineness of structure, recommendation is made that it be given an additional heat treatment by the manufacturers. In some cases a forging has been given two additional heat treat-

ments by the manufacturers before satisfactory results have been secured.

Whenever it seems advisable the transition range of a metal is investigated in order to determine the proper heat treatment for the forgings. Part of the apparatus for determining the transition range of a metal, by means of heating and cooling curves, is shown near the right of Fig. 1. It consists essentially of a furnace with thermocouples and a potentiometer for accurate measurement of temperatures. The cylindrical specimen used in determination of transition temperatures is about 1 inch long and $\frac{1}{4}$ to $\frac{1}{2}$ inch in diameter; it has holes drilled in it for the receipt of the thermocouples.

The simplest method of determining the transition range is by heating and cooling curves. For this purpose the specimen is slowly heated or cooled in the furnace, and temperature measurements are taken at definite intervals. A time-temperature curve may then be plotted. If the specimen has no transition points during this temperature range, a smooth, regular curve will be produced, as illustrated by Curve A of Fig. 2. If, however, there is a transition point, a sudden absorption or evolution of heat takes place at a definite temperature. This causes a lag in the temperature change and a sudden change in direction of the time-temperature curve. Curves B, C and D of Fig. 2 are forms of curves obtained with metal having a transition point.

One of the best methods of obtaining transition points consists in the use of an additional specimen of metal having no transition point within the required temperature range. The difference in temperature between the specimen investigated and this neutral specimen is then plotted against the temperature itself. On account of the lag in temperature change at the transition point, a peak in the difference curve occurs at this temperature, as illustrated by Curve E of Fig. 2. Several other forms of curves may be plotted, each having certain advantages and disadvantages.

After the transition range of a metal is determined or estimated, experiments in heat treatment are frequently made,

using an electric furnace and accurate temperature measurement. The results are determined by the use of the microscope and sometimes by physical tests. In drawing conclusions from these laboratory experiments, of course, due allowance must be made for the difference in size between the specimen and the forging from which the specimen was taken. Since, however, the difference in effects of heat treatment between a large forging and a small specimen depends almost entirely on their rates of cooling, allowance for this fact can readily be made.

Frequently methods of heat treatment have been suggested to manufacturers and have finally proved successful, while the methods originally used by the manufacturers have failed. The improvement often brought about by heat treatment suggested by this Station, is illustrated by Figs. 3 to 7. These are photomicrographs, at a magnification of 100, of typical specimens taken from 9-inch round bars of Class B steel. These specimens were etched with a 5 per cent. solution of picric acid in alcohol. The white areas represent ferrite, or nearly carbonless iron, while the dark areas represent pearlite, the carboniferous constituent of annealed mild steel. Figs. 3 and 4 illustrate the unsatisfactory structure found in these bars as first submitted by the manufacturers. In Fig. 3 streaked segregation is noticeable, while in Fig. 4 the ferrite is seen to be arranged in long, straight lines forming a triangular, latticework structure. In such material the coarse crystallization of the original ingot has not been completely removed.

Although a special heat treatment for this material was recommended by this Station, the manufacturers chose to try simple annealing, with the result that the structure of the material on second submission showed little improvement. This is illustrated by Figs. 5 and 6. When finally the heat treatment, originally suggested by this Station, was used by the manufacturers, a fine grained, uniform structure was produced, as illustrated by Fig. 7.

In the year and a half since metallographic requirements began to be included by the Bureau of Steam Engineering in

contracts for shafting, great improvement has been noticed in the structure of the material furnished by manufacturers. Although, from the nature of the case, metallographic specifications cannot be made as exact as tensile specifications, manufacturers have come to an understanding of the grade of metallographic structure that is considered acceptable. In only a small percentage of cases, therefore, is an additional heat treatment of the material required.

Occasionally it has been found necessary to examine a large object, such as a shaft of a battleship. For this purpose areas on the surface of the metal are polished and etched; they are then examined by means of the apparatus illustrated by Fig. 8. In this apparatus an aluminum arm attached to an aluminum base holds the microscope and camera. By means of the mechanism in the base the aluminum arm can be adjusted so that the optical axis of the microscope is perpendicular to the surface of the object to be examined. The objective of the microscope can be attached either in line with or at right angles to the main tube of the microscope. This gives a great variety of adjustment, and even makes it possible to examine the interior of a hollow object. A balopticon, such as the one shown in Fig. 8, is a very satisfactory form of illumination when an auxiliary condensing lens is used. The apparatus, however, could be equipped with an electric-light bulb and condenser rigidly clamped to the microscope.

The apparatus here illustrated has been used in examining a variety of large objects. Among other tests by this method a complete examination of 18 shafts for the U. S. S. *Pennsylvania* was made in 1914.

TESTS OF FAILED MATERIAL.

In addition to the various inspection tests, sometimes averaging over 20 in a month, this Station makes many tests of metals which have failed in service. These tests have included shafts, gun parts, rivets, stay bolts, boiler plate, pistons, valve stems, chain links, turbine blades and various articles of brass, bronze and other alloys.

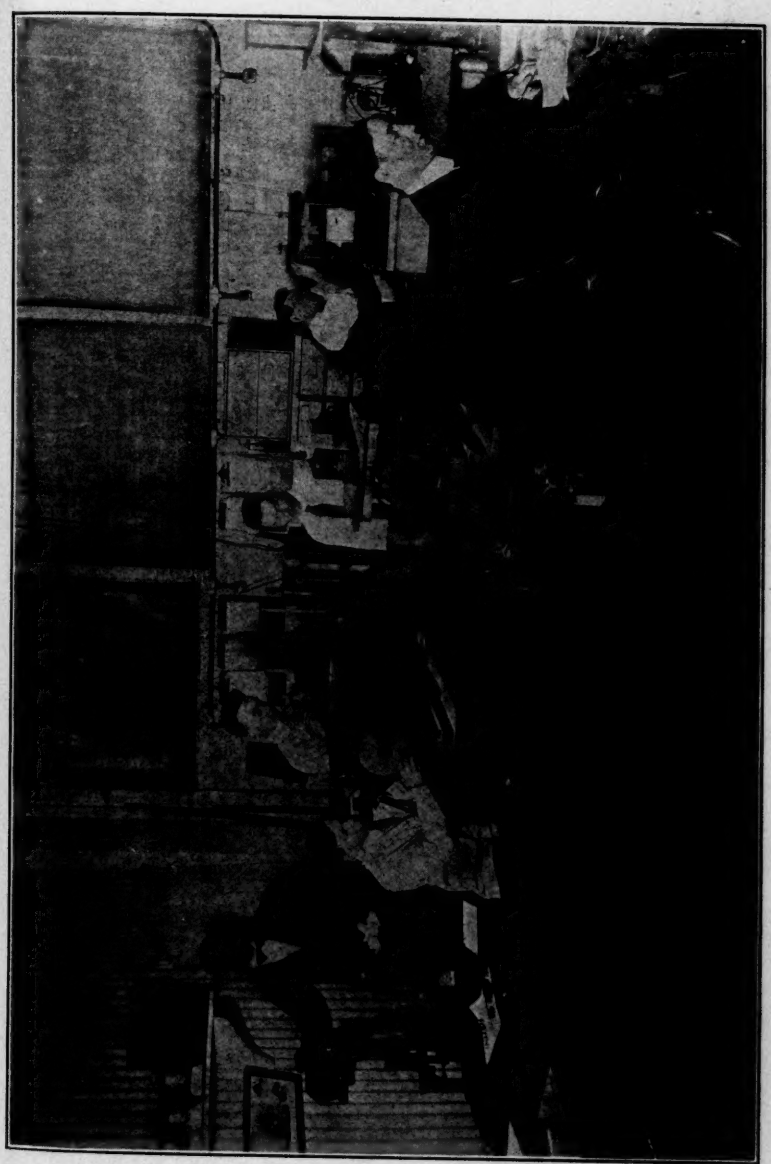


FIG. 1.

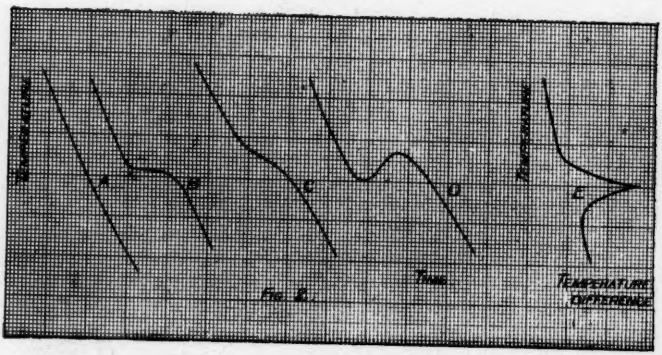

Temperature
A
B
C
D
E
Temperature
Time
Temperature Difference

Fig. 2.

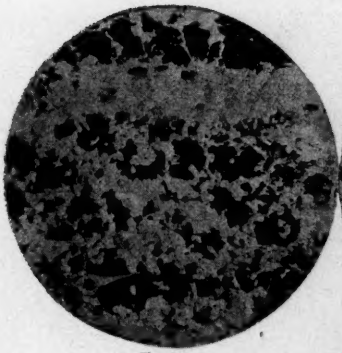


FIG. 3.

FIG. 4.

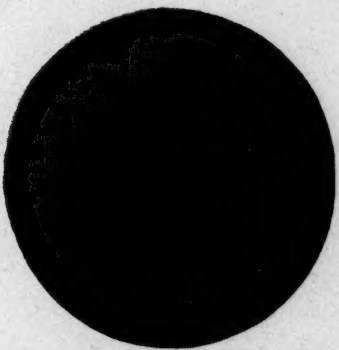


FIG. 5.

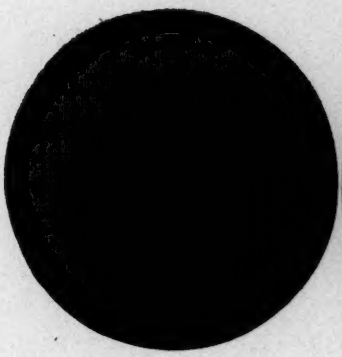


FIG. 6.

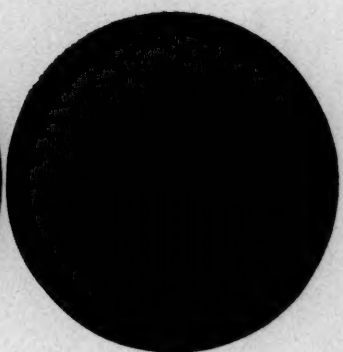


FIG. 7.

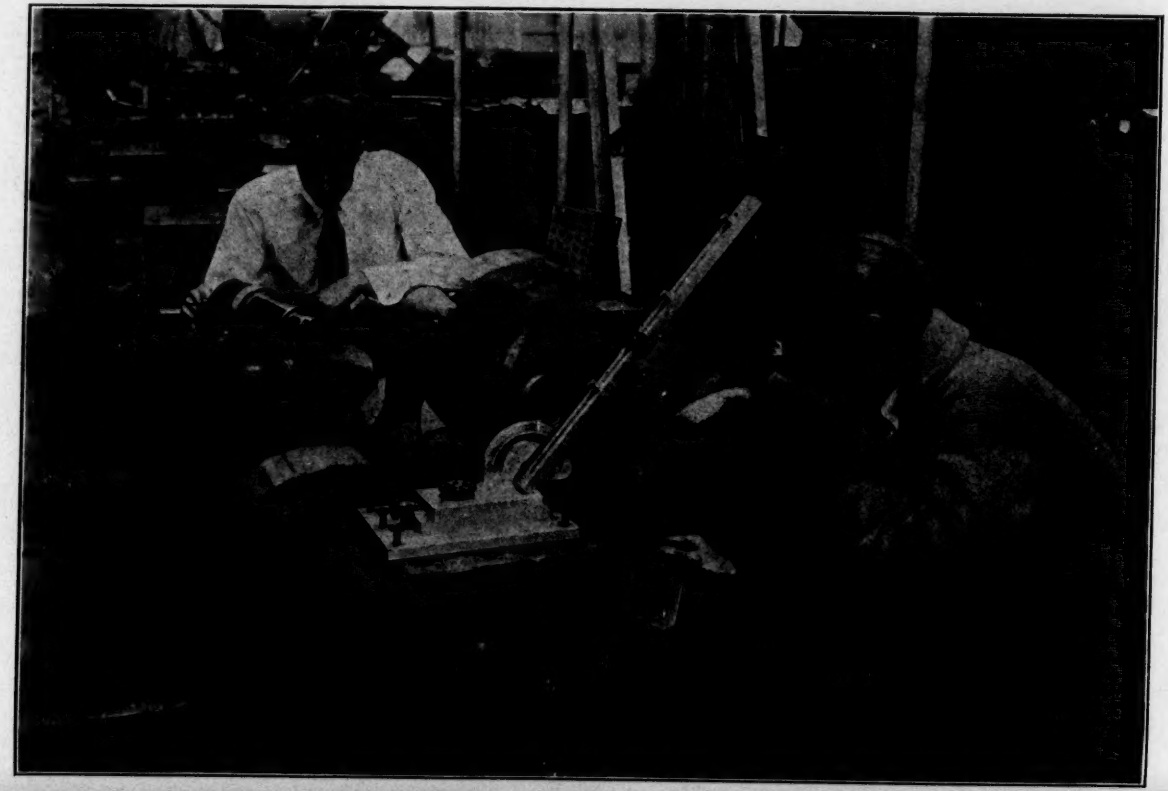


FIG. 8.

FIG. 9.

FIG. 10.

FIG. 8.

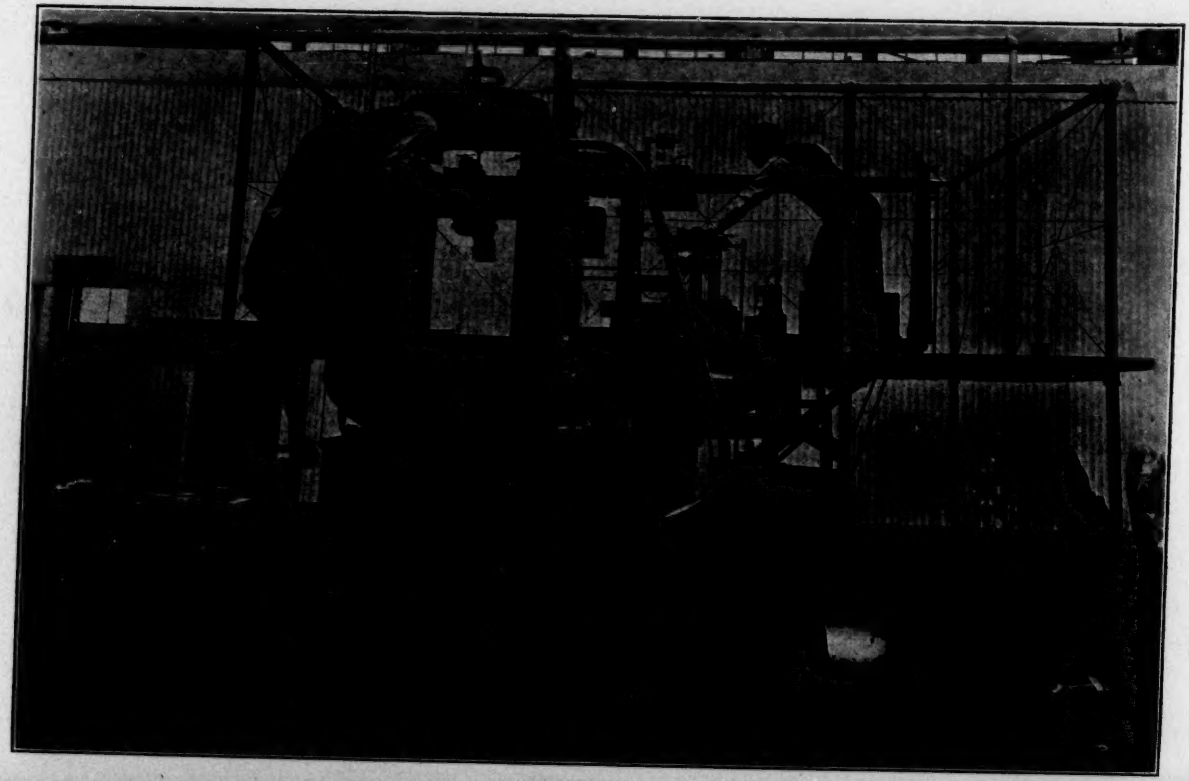


FIG. 11.

FIG. 11.

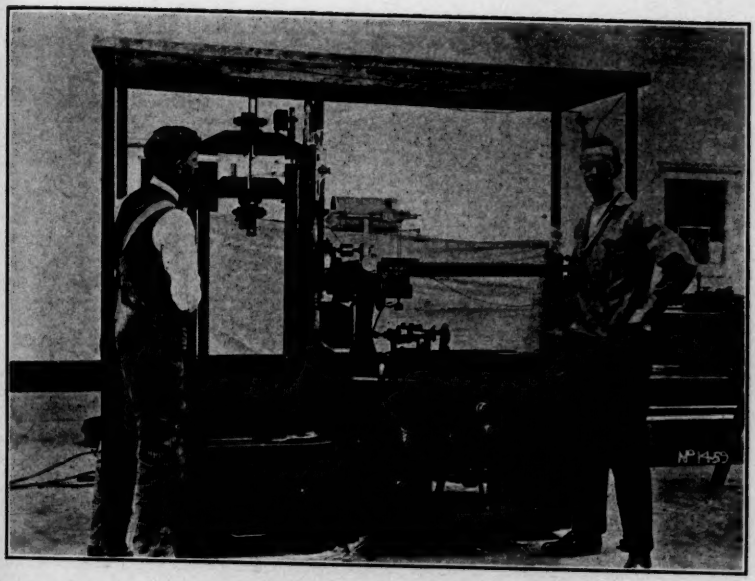


FIG. 12.

FIG. 13.

FIG. 14.

FIG. 15.

Figs. 9 and 10 show structures found in a failed crankshaft of submarine *G-1*. In Fig. 9 fine cracks extending through numerous slag particles are shown on the unetched surface. In Fig. 10 branching cracks are seen on the etched surface. Though other causes probably contributed to the failure of this material, there is no doubt that the metal itself was defective.

In investigating the causes of failure of metal it is, of course, necessary to keep in mind that a number of causes may have contributed to the result. In addition to defects in the metal itself, possible causes of failure are: faulty design, construction and alignment, also excessive stresses due to improper use. After examination of failed material, therefore, it is often impossible to decide definitely whether or not the failure was due to defective material. A decision can always be made, however, as to the quality of the metal.

PHYSICAL TESTING.

In investigating the quality of failed material, physical tests, in addition to the metallographic examination, are made whenever possible. The physical tests often include torsion, impact and endurance tests as well as the usual tensile tests.

At this Station the various machines for physical tests are situated in the central floor space. Figs. 11 and 12 are views of the two tensile machines having capacities of 300,000 and 50,000 pounds, respectively. Fig. 13 shows a Brinell machine for testing the hardness of metals. Fig. 14 is a view of a machine for testing by impact. A machine for testing the endurance of metals under alternate bending stress is shown in Fig. 15.

METALLURGICAL INVESTIGATION.

With these machines, in addition to routine tests, investigations are being made of the strength and endurance of metals and their ability to resist suddenly applied stresses. By means of these physical and metallographic tests a study is being

made of the relation between microscopic structure and physical properties. This subject, about which there is much disagreement among metallurgists, offers a wide field for investigation. The effects of varying heat treatment on both microscopic structure and physical properties are also being studied. Some of the results of this work will be described in a later article.

TESTS IN SERVICE.

In addition to all such artificial tests of metals the Engineering Experiment Station has the advantage of being in touch with tests for which all others are preliminaries, the tests of material in actual service in the Navy. The data in regard to microscopic structure, strength and endurance being accumulated at this Station will, therefore, in time prove of great value if considered in regard to the subsequent history of the various kinds of material in service. This fact alone illustrates the benefit to the Navy of having a central research laboratory such as the Engineering Experiment Station.

U. S. S. *CONYNGHAM.*

DESCRIPTION AND TRIALS.

By Lieutenant H. H. Norton, U. S. Navy, Member.

Torpedo boat destroyer No. 58, the *Conyngham*, is one of the six destroyers authorized by Act of Congress approved March 4, 1913. The contract for the building of this vessel, together with T. B. destroyer No. 59, the *Porter*, was awarded to the Wm. Cramp & Sons S. & E. B. Co., of Philadelphia, Pa. The contract was signed October 2, 1913, and 23½ months was the time allowed for completion. The contract price of the vessel was $881,000, of which $531,000 was allotted for machinery. The keel was laid July 27, 1914, and the vessel was launched July 8, 1915. The preliminary trials were held December 7 to 10, 1915, inclusive, and the vessel was delivered to the Government January 20, 1916.

Principal Hull Dimensions.

Length between perpendiculars, feet and inches	310–20
on L.W.L., feet and inches	310–00
overall, feet and inches	315–03
Beam, extreme, over guards, feet and inches	30–06½
on L.W.L., feet and inches	29–09½
molded, feet and inches	29–10
Draught, designed mean, feet and inches	9–04½
Displacement on designed draught, tons	1,090
per inch in L.W.L., tons	13.83

MACHINERY.

The propelling machinery consists of Parsons turbines arranged on two shafts as shown on Plate I. On the starboard shaft are L.P. ahead and astern turbines fitted in same casing. The cruising turbine shaft is geared to the forward end of the starboard shaft through a reduction gear and pinion of

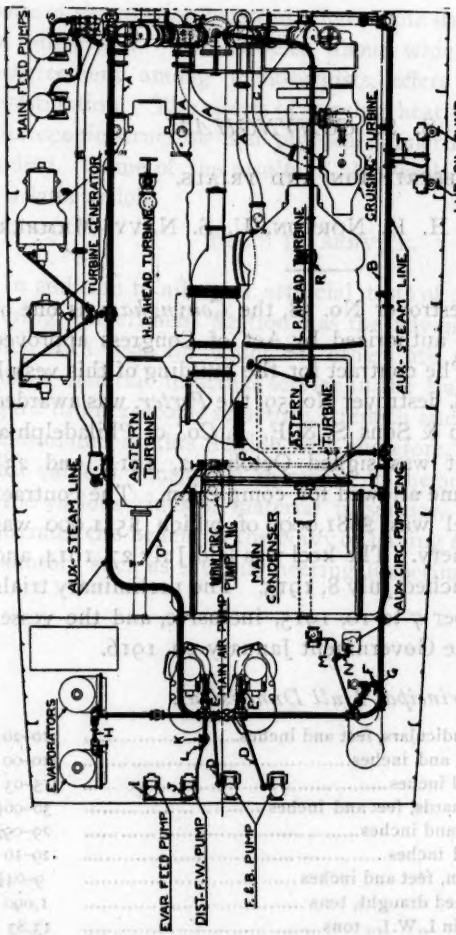

MAIN FEED PUMPS
TURBINE GENERATOR
H.P. AHEAD TURBINE
L.P. AHEAD TURBINE
CRUISING TURBINE
LUB. OIL PUMP
ASTERN TURBINE
AUX. STEAM LINE
MAIN CONDENSER
MAIN CIRC. PUMP ENG.
AUX. CIRC. PUMP ENG.
MAIN AIR PUMP
EVAPORATORS
EVAP. FEED PUMP
DIST. F.W. PUMP
F. & B. PUMP
A-STEAM TO AHEAD TURBINE
B-STEAM TO ASTERN TURBINE
C-STEAM TO CRUISING TURBINE
D-STEAM TO THR. & BLGE. PUMP AUX. ROOM
E-STEAM TO MAIN AIR PUMP
F-STEAM TO AIR COMPRESSOR AUX. ROOM
G-STEAM TO SEA CHEST AUX. ROOM
H-STEAM TO EVAPORATOR
I-STEAM TO EVAPORATOR FEED PUMP
J-STEAM TO DISTILLER FRESH WATER PUMP
K-STEAM TO STEERING ENGINE
L-STEAM TO HEATING SYSTEM
M-STEAM TO AUX. AIR PUMP
N-STEAM TO AUX. CIRCULATING PUMP ENGINE
O-STEAM TO MAIN INL. SEA CHEST
P-STEAM TO MAIN CIRC. PUMP ENGINE
Q-STEAM TO TURBINE GENERATOR
R-MAIN STEAM BLEEDER
S-STEAM TO MAIN FEED PUMP
T-STEAM TO LUBRICATING OIL PUMP

PLATE I.—U. S. S. "CONYNGHAM." ARRANGEMENT OF ENGINE AND AUXILIARY ROOMS.

ratio about 7 to 1. Between the reduction gear and L.P. turbine is fitted a Metten clutch, designed to uncouple the reduction gear from main turbine shaft when revolutions exceed a speed corresponding to 22 knots. On the port shaft are the H.P. ahead and astern turbines fitted in separate casings.

For cruising speeds up to 20 knots steam is admitted to

cruising turbine, then to first stage H.P. turbine, thence through L.P. turbine to condenser. For higher speeds ahead steam is admitted to first-stage H.P. turbine, thence through L.P. turbine to condenser.

For maneuvering, steam is admitted direct to L.P. ahead or astern turbines, H.P. ahead or astern turbines, and exhausts direct from turbine in use to condenser. Non-return valves are fitted between cruising and H.P. turbine and between H.P. and L.P. turbine. Between the H.P. and L.P. turbine a large direct-exhaust valve to condenser is fitted, to allow exhaust from H.P. turbine to pass direct to condenser when H.P. turbine is used for maneuvering. The main turbines are designed to develop about 9,000 S.H.P. each, when making about 600 r.p.m.

Turbine Data.

Diameter of cylinder, inches.				Length of cylinder, inches.				Rows of blading.				Length of blades, inches.			
Ahead.		As.		Ahead.		As.		Ahead.		As.		Ahead.		As.	
C.	H.P.	L.P.		C.	H.P.	L.P.		C.	H.P.	L.P.		C.	H.P.	L.P.	
$20\frac{7}{16}$	: $50\frac{1}{2}$	: 73	: 45	$9\frac{3}{8}$	: $14\frac{1}{2}$	: $13\frac{1}{2}$	: $8\frac{1}{4}$	10	: 11	: 7	: 7	$\frac{1}{2}$	: $1\frac{1}{2}$	: $6\frac{1}{2}$	: $0\frac{1}{2}$
$20\frac{1}{2}$	: $51\frac{1}{2}$	: 79	: 46	11	: $16\frac{3}{16}$	: $15\frac{1}{2}$	: $10\frac{1}{2}$	10	: 11	: 7	: 7	$\frac{3}{4}$	: $1\frac{3}{4}$	: $9\frac{1}{2}$	: 1
$21\frac{1}{2}$	: $52\frac{1}{2}$	: 86	: 48	$13\frac{1}{2}$	: $15\frac{1}{2}$	: $15\frac{1}{2}$	: $10\frac{3}{8}$	10	: 11	: 6	: 7	$1\frac{1}{2}$	: $2\frac{1}{2}$	: 13	: 2
	: $54\frac{1}{2}$	: 86	: 51		: $16\frac{7}{16}$	: $15\frac{3}{8}$	: $9\frac{1}{2}$		: 11	: 6	: 5		: $3\frac{1}{2}$	: 13	: $3\frac{1}{2}$
	: 57	: 86	: 51		: $18\frac{1}{2}$	: 18	: $7\frac{1}{2}$		: 10	: 6	: 4		: $4\frac{1}{2}$	: 13	: $3\frac{1}{2}$
	: 61	: ...	: 51		: $21\frac{3}{8}$	: ...	: 4		: 10	: ..	: 2		: $6\frac{1}{2}$	: ...	: $3\frac{1}{2}$

Diam. of rotor drums: C 1st Exp. $19\frac{7}{16}$ ins.; 2d and 3d Exps, 19 ins.; H.P. 48 ins.; L.P. 60 ins.

Standard Parsons material for blades and method of securing them is used. Radial dummy packing is used on L.P. ahead and both astern turbines. Contact dummy packing is used on H.P. and cruising turbines; only these latter two being fitted with micrometers for reading dummy clearances.

Auxiliary exhaust connections to first and second stages are fitted on the H.P. turbine. This turbine is also fitted with a by-pass connection from first to second stages. Auxiliary exhaust connections to the first and second stages are fitted on the L.P. turbine.

Shafting and Bearings.

H.P. rotor shaft, length, feet and inches, forward	5–11½
aft	4–01 1/16
diameter at bearings, inches	10
of hole, inches	05 and 06
L.P. rotor shaft, length, feet and inches, forward	6–03⅜
aft	5–02⅞
diameter at bearings, inches	11
of hole, inches	5 and 05¼
Line shaft, length, starboard, feet and inches	19–10
	19–10
port, feet and inches	18–00
	19–10
diameter at bearings, inches	09½
of hole, inches	06½
Stern-tube shaft, length, feet and inches	22–03
diameter, inches	09⅜
diameter of hole, inches	06½
Propeller shaft, length, feet and inches	26–08 5/16
diameter, inches	09⅜
diameter of hole, inches	06½
Rotor shaft bearings, number of, each turbine	2
length, H.P. forward, inches	12½
after, inches	12½
L.P. forward, inches	20¼
after, inches	20¼
cruising, forward, inches	07
after, inches	07

Collars on shaft:	C.T.	H.P.	L.P.
Number	8	17	11
Thickness, inch	00⅜	00 1 1/16	00 1 1/16
Space between, inch	00 9/16	00⅞	00 1 3/16
Outside diameter, inches	07	13	13
Inside diameter, inches	04	08½	08½
Bearing surface, square inches	189	1,129	706
Thrust shoes, number, ahead	8	.....	.....
astern	7	16	10
effective (thrust) surface square inches, ahead	86	546	686
effective (thrust) surface square inches, astern	75	552	666
Line shaft bearings:			
Number of bearings and length, ins., starb'd	3	.....	16
port	2	.....	16
Stern tube bearings:			
Length, forward bearing, inches	.....	.....	27
after bearing, inches	.....	.....	43
Propeller shaft bearings:			
Length, strut bearing, inches	.....	.....	54 3/16

Thrust bearings are located one at the forward end of H.P., one at forward end of L.P. turbine, and one at the after end of cruising turbine.

Gary-Cummings torsion meters, one for each line shaft, are installed for measuring S.H.P.

Propellers.

One three-bladed outboard-turning propeller is fitted on each shaft. Propellers are cast solid of manganese-bronze, and blades are true screw and machined to pitch.

The following are actual propeller measurements:

Diameter, feet and inches	7–10
Pitch, as set, mean, feet and inches	6–10
Ratio of diameter to pitch	1 : .872
Area, projected, square feet	28.92
Area, helicoidal, square feet	32.31
Area, disk, square feet	48.19
Depth of lower tip of blade below keel, inches	01½
Immersion of upper tip of blades (at draught 9′4½″) ins	20

Condensers.

There is but one main condenser, which is located athwartship in after part of engine room. It is of a cylindrical shape with tubes curved horizontally and rolled into tube sheets. Tubes are supported at an intermediate point between tube sheets, but not secured to support, play being allowed for expansion.

One auxiliary condenser, located in auxiliary room, is installed. It is similar in type to main condenser. It is equipped with an independent air pump and independent circulating pump.

Main Condenser.

Tubes, number	6,351
thickness, B.W.G	18
outside diameter, inch	00⅝
length, as fitted, feet and inches	11–05 9/32
cooling surface, square feet	11,610

Auxiliary Condenser.

Tubes, number	291
thickness, B.W. G.	18
outside diameter, inch	00⅝
length, as fitted, feet and inches	6–06
cooling surface, square feet	302

Engine-Room Auxiliaries.

Feed and Filter Tank.—A feed and filter tank of usual type, having a capacity of 800 gallons, is located on port side of auxiliary room.

Pumps.

See Table I.—List of pumps with their connections.

TABLE I—PUMPS AND CONNECTIONS—U.S.S. CONYNGHAM.

No.	Pumps	Size (ins.)	Type	Suction Pipes From		Discharge Pipes To		Location
				Ins.		Ins.		
2	Main Air	14 x (2) 20 x 18	Twin, Vert., Beam Bucket, "Blake."	11	Condensers thru water seal.	9	Feed Tank.	Aux. Room.
1	Main Circulating	11 x 9 Eng. 39 Runner	Engine-Driven, Double-Inlet, Cent., "Cramp."	24 10	Sea. Bilge.	24 3	Main Condenser. Oil Cooler.	Eng. Room.
2	Main Feed	15 x 10 x 16	Vert., Simplex, Piston, "Blake."	5½ 5½	Feed suction pipe & reserve feed tanks. Water seal cross-connecting pipe.	4	Main feed discharge.	Do.
2	Aux. Feed	Do.	Do.	5½ 1½	Feed suction pipe & reserve feed tanks. Hose connection.	4 1½	Auxiliary feed. Hose connection.	One in each fire-room.
4	Fire & Bilge.	7 x 7 x 12	Do.	4 4 1½	Sea. Drainage. Hose connection.	2½ 2½ 2½ 1½	Fire main. Overboard. Distillers & flushing system thru fire main. Hose connection	Two in Aux. Room. One in each fire-room.
1	Aux. Air	4½ x 8 x 6	Do.	4	Aux. Condenser.	3	Feed Tank.	Aux. Room
1	Aux. Circulating	9½ x 9½ Eng. 23 Runner	Engine-Driven, Cent., "Cramp."	4	Sea.	4	Aux. Condenser.	Do.
1	Evaporator Feed	4½ x 6 x 6	Vert., Simplex, Piston, "Blake."	2 1½	Sea. Distiller circulating water disch.	1½	Evaporators.	Do.
1	Distiller Fresh Water.	Do.	Do.	2½	Distiller reservoir tank.	2 2 2 2	Reserve feed tanks. Ship's tanks. Cofferdam. Main feed tank.	Do.
1	Oil to Clutch.	3 x 1¾ x 4	Do.	¾	Lubricating oil discharge line.	¾	Oil Clutch.	Eng. Room.
2	Lubricating Oil	4½ x 6 x 6	Do.	3	Lubricating oil tank.	2½ 2½	Bearings. Settling Tanks.	Do.
4	Fuel Oil Service	6 x 3½ x 8	Vert., Duplex, Piston, "Blake."	2½ 2½	Oil Storage Tanks. Booster Pump Disch.	2	Burners.	Two in each fire-room.
2	Fuel Oil Booster	4½ x 6 x 6	Vert., Simplex, Piston, "Blake."	3 3	Storage Tanks. Deck-Hose Conn.	2½ 2½ 2½	Service Pump Supply. Storage Tanks. Deck-Hose Conn.	

FORCED-LUBRICATION SYSTEM.

This system, comprising two lubricating-oil pumps and one oil cooler of 210 square feet cooling surface, supplies forced lubrication to all main bearings, thrust bearings, main circulating engine and reduction gear.

The cooling water for oil cooler is supplied by main circu-

lating pump. A connection to firemain is available for an emergency. The main circulating engine has additional forced lubrication supplied by a plunger pump driven from eccentric strap.

The Metten clutch has its oil pressure supplied by a small plunger pump driven by reduction-gear shaft. For emergency, the forward one of the main lubricating, or a small auxiliary oil pump located on forward engine-room bulkhead may be used.

One oil-drain tank of 205 gallons capacity, two oil-settling tanks of 200 gallons capacity each, and main storage tanks of 550 gallons capacity, are comprised in this system.

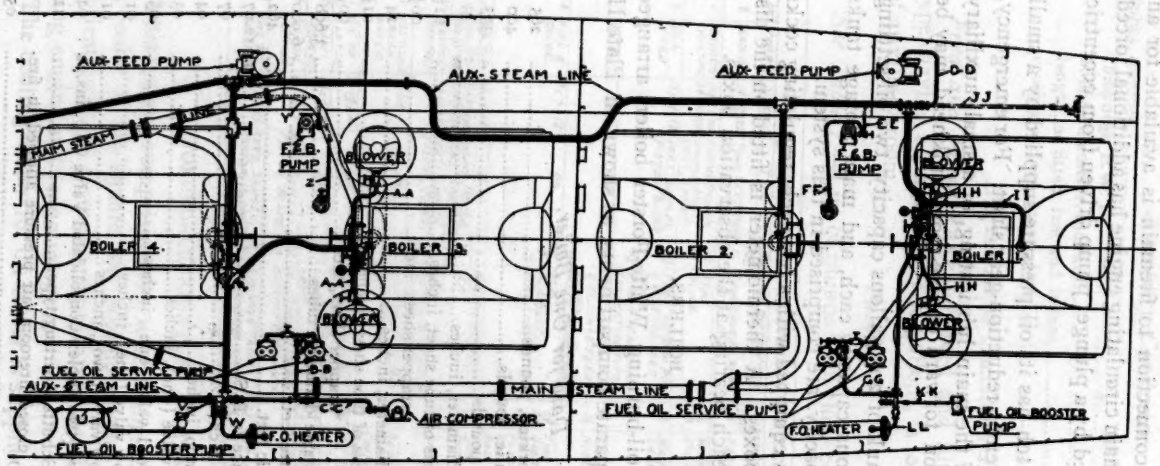
Each bearing is equipped with a test cock, regulating cocks and observation boxes. A thermometer is fitted on the discharge line from each bearing at the observation boxes.

BOILERS.

There are four oil-burning White-Forster boilers arranged in pairs in two separate compartments, as shown in Plate II.

Data for One Boiler.

Pressure, designed working, pounds	265
test, hydraulic, pounds	400
test, steam, pounds	335
Drum, steam, inside diameter, inches	48
thickness of tube sheet, inches	01 11/16
wrapper sheet, inch	00 11/16
water, inside diameter, inches	24
thickness of tube sheet, inches	01 1/2
wrapper sheet, inch	00 5/8
Number of 1-inch, outside diameter, tubes	3,608
Heating surface, square feet	6,000
Furnace volume, cubic feet	462.5
Ratio combustion space to H.S.	1 : 12.97
Diameter, main stop, inches	07
auxiliary stop, inches	04
main feed valve (one), inches	03
auxiliary feed valve (one), inches	03
surface-blow valve (one), inches	01 1/2
bottom-blow valve (two), inches	01 1/2
safety valve (two duplex Consolidated Valve Co.), ins.	04
Number of burners, Schutte-Koerting type	11
Kind of forced draft, closed fireroom, air pressure allowed in fire-rooms, inches	05



U-STEAM TO LUBRICATING OIL SETTLING TANK
V-STEAM TO FUEL OIL BOOSTER PUMP
W-STEAM TO FUEL OIL HEATER
X-STEAM TO AUX-FEED PUMP AFT-FIRE ROOM
Y-STEAM TO FIRE & BILGE PUMP AFT-FIRE ROOM
Z-STEAM TO SEA CHEST AFT-FIRE ROOM
AA-STEAM TO BLOWER TURBINE AFT-FIRE ROOM
BB-STEAM TO FUEL OIL SERVICE PUMP AFT-FIRE ROOM
CC-STEAM TO AIR COMPRESSOR AFT-FIRE ROOM

DD-STEAM TO AUX-FEED PUMP FORD-FIRE ROOM
EE-STEAM TO FIRE & BILGE PUMP FORD-FIRE ROOM
FF-STEAM TO SEA CHEST FORD-FIRE ROOM
GG-STEAM TO FUEL OIL SERVICE PUMP FORD-FIRE ROOM
HH-STEAM TO BLOWER TURBINE FORD-FIRE ROOM
II-STEAM TO WHISTLE
JJ-STEAM TO DECK MACHINERY
KK-STEAM TO FUEL OIL BOOSTER PUMP FORD-FIRE ROOM
LL-STEAM TO FUEL OIL HEATER FORD-FIRE ROOM

PLATE II.—U. S. S. "CONYNGHAM." ARRANGEMENT OF FIRE ROOMS.

Each boiler fitted with independent smokepipe—height above main deck 20 feet.

All tubes are curved of variable radii of 5 feet 6½ inches to 6 feet 9⅛ inches, with the smaller radii used for outer and inner rows.

These boilers are similar in type and construction to previous White-Forster boilers, except that the design of mud drums has been changed from a semi-elliptical to a cylindrical shape. This necessitated a change of design in front head of steam drum to facilitate the removal of tubes. To accomplish this, in place of two manholes of equal size, as originally used, there is but one large manhole, 12 inches by 16 inches near top of steam-drum head, and eight small cylindrical manholes, 2½ inches in diameter, distributed in two rows across the center of drum head. Butt seams are used on both steam and mud drums.

FUEL-OIL SYSTEM.

This system comprises two booster pumps, one in each fireroom; four service pumps, two in each fireroom; two hand pumps, one in each fireroom; two fuel-oil heaters, one in each fireroom.

The booster pumps are used principally for handling oil stowage, though they may be used to assist fuel-oil service pumps, the suction of service pump being connected to discharge of booster pump.

The hand pumps are installed for use when no steam is on ship. They discharge direct to burner line.

The fuel-oil service pumps discharge through oil heater to burner line. Provision is made for by-passing oil heater.

In each fireroom the discharge line to burners is fitted with a stop valve controlled from deck. Each burner line, 1 5/16 inches in diameter, is fitted with a stop cock and automatic shut-off valve. Branches ⅜-inch in diameter supply the eleven burners.

Strainers are fitted in both suction and discharge side of

service pumps. Air chambers are fitted on discharge side of service pumps.

FORCED-DRAFT BLOWERS.

Forced draft is supplied to each fireroom by two 35-inch Keith-Sturtevant fans direct connected and driven by Terry turbines.

FEED-WATER HEATER.

There is one horizontal surface feed-water heater, 6 feet 7⅛ inches in length and 17 inches diameter, installed in each fireroom. Each heater has 238 tubes of ⅝-inch diameter, fitted with spiral retarders, and have a total heating surface of 250.08 square feet.

MAIN STEAM PIPING.

The main steam connections on each boiler are 7 inches in diameter. These connections from forward boilers combine into a 9½-inch line and lead aft on starboard side to a main stop valve at forward engine-room bulkhead.

In a similar manner the after boiler connections combine and lead aft on port side to a main stop valve. From these stop valves 9½-inch lines lead to the turbine-throttle manifold, through which these lines are connected, there being no separate cross-connection piping.

An expansion joint is fitted in starboard line in forward fireroom, and in the port line in the after fireroom.

The throttle manifold comprises a 13½-inch globe stop valve controlling steam to H.P. turbine, a 6½-inch globe stop valve controlling steam to cruising turbine, and two ahead and astern maneuvering manifolds. A maneuvering manifold comprises two 6-inch balance poppet valves dually operated by a screw spindle through a spring-loaded rocker beam.

AUXILIARY STEAM PIPING.

Auxiliary steam connections on each boiler are 4 inches in diameter. The connections from the forward boiler unite on the port side and lead aft to after fireroom.

After connecting with the boilers in the after fireroom the line is branched and led aft on each side of ship through engine room with a loop at the after end of auxiliary room. From these lines, branches are led in the various compartments to all auxiliaries.

AUXILIARY EXHAUST PIPING.

The auxiliary-exhaust piping runs throughout machinery spaces. There are branches to each auxiliary, and the following connections.

8 inches to main condenser.

6 inches to auxiliary condenser.

6 inches to 1st and 2d stages H.P. turbine, 1st and 2d stages L.P. turbine.

7 inches to each feed-water heater.

3½ inches to turbine glands.

MAIN AND AUXILIARY FEED SYSTEMS.

There is a 7½-inch suction main running forward from feed and filter tank with a 5½-inch connection to air-pump channel ways and a 5½inch connection to each main feed pump in the engine room. The line leads forward through 7-inch pipe to after fireroom where a 5½-inch connection is taken off to auxiliary feed pump, then continues forward through 5½-inch pipe to auxiliary feed pump in forward fireroom.

The discharge from each feed pump is 4 inches in diameter, the discharges from two main feed pumps combining into a 5½-inch line which leads forward with a 4-inch branch to each feed-water heater or by-pass, thence through a 4-inch line to steam drums of the boiler.

Each auxiliary feed line discharges direct through 4-inch line to steam drums or cross-connects to main feed line through heater or by-pass.

EVAPORATING AND DISTILLING PLANT.

This plant comprises two evaporators, located on port side of auxiliary room, one evaporator-feed pump and one fresh-water pump, one evaporator-feed-water heater and two distillers.

This plant has a normal combined capacity of 5,400 gallons of fresh water per 24 hours, with a 40 per cent. excess of normal capacity when clean.

Data for One Evaporator.

Type	Schutte-Koerting, Navy type, vertical, U-tube.
Diameter, inside, inches	28
Height overall, feet and inches	5–11⅛
Number of U-tubes	36
Diameter of tubes, inside, inches	01½
Heating surface, square feet	59.61
Diameter steam nozzle, inches	02
vapor nozzle, inches	05
feed valve, inch	01
blow valve, inches	02

Data for One Distiller.

Type	Cramp, vertical, curved tube.
Diameter, inside, inches	13
Height between covers, feet and inches	2–10½
Number of tubes	100
Diameter of tubes, outside, inch	00⅝
Cooling surface, square feet	45
Diameter vapor inlet, inches	03½
drain connection, inches	01½
circulating-water connections, inches	02

Data for Evaporator Feed-Water Heater.

Type	Cramps, vertical, U-tube.
Diameter, inside, inches	11½
Height overall, feet and inches	3–08 9/16
Number of tubes	9
Diameter of tubes, outside, inch	01
Heating surface, square feet	14.24
Diameter vapor connections, inches	05
feed-water connections, inches	01½

SPECIAL FEATURES.

Throttle Locking Gear.—To prevent possible injury to the oil clutch by admission of steam to backing turbines while the cruising turbine is in use, or vice versa, the cruising-turbine throttle and maneuvering throttles are interlocked in such a manner that as soon as any of the maneuvering tur-

bine throttles are opened the cruising-turbine throttle is locked, or, should the cruising-turbine throttle be opened, the maneuvering throttles would be locked. This is accomplished through a system of cranks and connecting links operated direct by the operating valve spindles, so that the first movement toward opening a throttle valve immediately locks the other throttle valves, whose accidental opening would be injurious to the installation.

To prevent the simultaneous opening of cruising and H.P. turbine throttles, which would cause the exhaust from cruising turbine to back up and close the non-return valve, the throttle-valve stem of the H.P. turbine is incorporated in the above locking system.

Hydraulic Operating Gear for H.P. Turbine Exhaust Valve.—To facilitate the operation of the 30-inch H.P. turbine receiver valve controlling exhaust from H.P. turbine, a hydraulic cylinder is attached to the operating spindle. This cylinder derives its power at either end through connections to the main feed-pump discharge line. The direction of applied power is controlled through a four-way cock directly connected to an operating lever located at throttle manifold. Two passages of this four-way cock are connected to main feed discharge line, and the other two passages to the main feed suction line. Thus power is applied to either side of piston in hydraulic cylinder and at the same time any pressure on the other side of piston is released through main feed suction line. To prevent throttle valve for H.P. turbine combination being opened except when exhaust from H.P. turbine is opened to L.P. turbine, a locking gear is fitted at the end of operating lever which engages with H.P. turbine throttle-valve spindle when exhaust to L.P. is closed.

Smoke Indicator.—A direct-reading smoke-indicator system, as manufactured by the Pioneer Smoke Indicator Company, Philadelphia, Pa., was installed for experimental purposes.

The principle of operation of the system is dependent on the property of selenium to vary its electrical resistance with

varying light intensities, viz: of low resistance at higher light intensities and high resistance at lower light intensities.

Use is made of this principle by projecting across the smokepipe a strong beam of light on a selenium cell. As the density of the smoke passing through the stack varies, the intensity of the projected light beam will also vary, which in turn will vary the electrical resistance of the selenium cell. The selenium cell is connected to an indicating meter having its scale arranged in smoke densities. The meter, being located in the fireroom, gives constant information to those operating the boilers as to the character of smoke.

This system, in use during the official trial, was reported as operating most satisfactorily.

TRIALS.

The standardization trials, 26 runs being made, were conducted on December 9, 1915.

The official speed and revolution curve gave the following mean revolutions per minute of the propellers, as necessary to attain the speeds stipulated by trials:

Speed, in knots.	R.p.m.
12,	190.4
15½,	246.0
16,	253.6
20,	319.7
25,	433.2
29.5,	580.8

THE INDUSTRIAL AND STRATEGICAL IMPORTANCE OF OUR NAVAL STATIONS.

By Rear Admiral John R. Edwards, United States Navy (Retired), Member.*

"To Prepare a Fleet is Vitally Important, To Take Care of a Fleet and Repair It is Equally Important."

INTRODUCTORY.

If the United States should become involved in war with a formidable naval foe the first guiltless victims to have their professional reputations blighted would be some of the naval officials responsible for the administration of our naval shore stations.

The first links in the chain of naval operations to be subjected to strain in preparing the fleet for sea would be the navy yards, and, if they failed under war requirements, would not the Nation be justified in demanding that someone be held responsible for such weakness; for has not the doctrine been incessantly proclaimed, even by naval experts,

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that we not only have too many navy yards, but that too much money has been spent upon them?

So long as this Nation's commercial resources, as regards the repair and construction of vessels of very heavy tonnage, are of such limited nature it will be impossible to obtain naval preparedness, except at inordinate expense and loss of time. The events of the past year have established the fact that our next great war will be fought out in the shop and the mill as well as in the trench and the turret.

Our commercial shipyards and naval stations have been continuously discredited; their development has been arrested; and, therefore, they can not satisfactorily do the work they will be called upon to perform.

The insufficiency of storehouse facilities at our various naval stations, the limited extent of deep-water piers, marked lack of dry docks, and the incomplete equipment of modern cranes would necessitate such serious delay in preparing the fleet for sea that the resentment of the Nation would be justly aroused and the demand made that some administrative officials should be held accountable for the inexcusable lack of preparedness of the shore establishments for fulfilling the purpose for which they were established.

THE PROBLEM OF THE NAVY YARD IS VITAL TO NAVAL EFFICIENCY.

The problem of the navy yard is an important one, both from an industrial and military standpoint. It seems inexplicable that there could exist in the service such incomplete and misleading information as that which prevails concerning the character and extent of the equipment of our navy yards. Should it not have been recognized long ago that well-equipped shore stations are essential to the efficient operation of the battle fleet?

It is regrettable that so many able officers and so many intelligent people should believe that we possess any unnecessary naval stations. It is especially unfortunate that this

opinion should be held by a number of influential naval officers now holding responsible positions at the Navy Department, particularly as these officers to a considerable extent are enabled to have their views enunciated as the policy of the Nation.

The possibility that the Navy Department and the Congress may indorse the erroneous belief that our existing naval shore establishments can meet the war needs of the fleet should cause grave concern to the Navy at large. Our administrative officials can not afford to be wrong as regards the problem of the navy yard.

THE RELATION OF THE NAVY YARD TO THE FLEET.

The navy yard is something more than a ship construction and repair establishment. They should be regarded as vitally component parts of the fleet.

As bearing upon the question of how navy-yard development should be effected, attention is invited to an extract from a report by the Naval Board of Inspection for Shore Stations, made in 1913, wherein it is stated:

> "It is of paramount military importance to the efficiency of the fleet that coincident with its extension there should be a corresponding development of our leading navy yards and stations. Every additional battleship authorized by Congress entails a corresponding military responsibility or obligation to provide means for its efficient upkeep as well as for its rapid and efficient repair. It is therefore imperative that the navy yards should be developed along every line which would fit them for more readily meeting the demands of the fleet.
>
> "Any delay in building up the navy yards or any expectation of depending primarily upon private shipbuilding plants for meeting war demands of the fleet impairs naval efficiency and is a menace to our first line of national defense. The danger of such a policy

may not be appreciated in time of peace, but there will come peril, if not national humiliation, if navy-yard development is unduly restricted.

"An extensive array of auxiliary vessels is the concomitant of an efficient fleet. Consistent likewise with an increase in number of battleships and naval auxiliaries, it is equally essential that there be a parallel development of navy-yard facilities."

OUR MARITIME FACILITIES FOR "REPAIR" SHOULD BE CLOSELY ASSOCIATED WITH THE ABILITY TO "PREPARE."

It is inviting disaster to "prepare" without providing facilities for "repair." Almost invariably defeat has awaited the commander who, throwing his all into the first line of battle, failed to provide a reserve in which to re-form and refit. The overshadowing advantage of Germany in this war has been due to her ability to repair and refit her battered battle lines. The call for adequate facilities for "repair" should be placed upon the same plane as the nation-wide demand to "prepare." Is it not the ability to put steam into the punches of the closing rounds of the fight that often decides the battle? If our naval stations are not in condition to make rapid and adequate repairs, should not development of these stations be immediately undertaken along with the extension of the fleet?

HAS THERE BEEN EXCESSIVE WASTE IN NAVAL EXPENDITURES?

Does it contribute to the efficiency of the Navy that one can not even cursorily read current literature without the conviction that even high ranking officers of the service stand sponsors for the statement that there has been extravagance and waste in the expenditures that have been made for the maintenance and development of our naval stations? No reliable evidence has been presented to materially sup-

port this contention, although some minor wastes and mistakes might be pointed out.

It is difficult to understand what purpose was intended to be subserved by the dissemination of views which impeach the ability of those naval officers responsible for the establishment and maintenance of our naval stations. Every unjust implication about the naval shore stations reflects injuriously upon the Navy as a whole, since such criticisms are certain to be misunderstood by the press at large and by the Naval Committees of the Congress. All such criticisms are a condemnation of the manner in which we have been administering our navy yards, and they will surely be regarded by the casual reader as applicable likewise to the general administration of the Navy.

It is unjust to attempt to ascribe any excessive cost of maintenance of the Navy to waste at the shore establishments, when the total expenditure for labor at all our naval stations is only $22,000,000, or about 16 per cent. of the total amount appropriated by the Congress for naval purposes. The fleet can not be built up by discrediting the navy yards, since the total annual expenditures for labor and material at all the shore stations would not be sufficient to pay for two modern superdreadnaughts.

The efficiency of the navy yards will compare favorably with that of any other feature connected with the operation of the fleet, since there is an interchange of naval officers in performing duty ashore and afloat. The same zeal, high purpose, and ability manifested in the performance of duty afloat is rendered by the sea-going officers when detailed for shore duty. Impeaching the character of the work done at the naval stations is simply discrediting that done afloat.

MISLEADING STATEMENTS CONCERNING NAVAL EXPENDITURES.

In one of the many characteristic magazine articles written about our navy yards it was stated that the naval expenditures for the fiscal year 1914 exceeded $140,000,000, and

that with this sum (the greatest peace appropriation for naval purposes in our history) only one ship of the first line was provided for. As a matter of fact that particular naval appropriation bill, the act of March 4, 1913, containing an appropriation of $35,325,695 for increasing the Navy, not only provided for the construction of a battleship but likewise for six destroyers and four submarines, together with a transport and a supply ship.

The implication was also made in this particular contribution that certain expenditures for naval purposes were of a character that made it difficult to satisfy reasonable men as to their necessity. It is certainly remarkable that anyone can carefully analyze any annual naval appropriation bill, propound questions of the character contained in that article, and then formulate conclusions that are not only injurious to the Navy but which have not stood the test of even two years' study of the matter.

IMPEACHMENT OF THE EFFICIENCY OF OPERATION OF OUR NAVAL SHORE STATIONS.

The sea power of this Nation can never be built up by unjust attacks upon the naval shore stations. The adequate development of the navy yard is the corollary to the increase of the fleet. Despite the extent to which our shore stations have been discredited, the efficiency of the navy yards is as high as that of the fleet. Continued attacks upon the navy yard will result in impeaching the work done afloat.

The efficient performance of duty at our naval stations requires extended professional experience, mature judgment, tact, integrity, and executive ability. Why, therefore, should continued effort be exercised to discredit this work. In justification of the attacks that have been made for the past generation upon navy-yard administration, our legislative and executive department officials have even been informed that an annual savings could be effected, whereby there could be obtained funds for the building of a battleship. The con-

tention that it is possible each year for a $15,000,000 battleship to be constructed from savings that would result from more economical disposition of the $22,000,000 now annually expended for labor at our navy yards is but in line with equally unwarranted implications made by various naval writers, who have unintentionally but unjustly discredited the work conducted by their fellow officers assigned to duty at our naval stations.

Where condemnation of our policy of navy-yard development is practically unqualified, the service at large has a right to inquire as to the extent and character of the industrial duty performed by these critics, and also as to their qualifications for passing judgment upon this question. It will be difficult to find any who have had extended or important engineering duty at any naval shore station who will indorse statements of the following character:

> "That our navy yards as a whole represent the aspect of a makeshift; that they conduce to waste of public funds; and that they represent expediency of sectional interest without any comprehensive, well-digested plan looking to the best good of the whole Navy."

RELATIVE NAVAL STRENGTH CAN NOT BE WHOLLY MEASURED BY COMPARATIVE STRENGTH IN BATTLESHIPS.

Considering the naval events of the past year, many of the leaders in the Congress upon whom has been delegated the responsibility of directing legislation will not be inclined to believe that relative naval strength should be measured solely in terms of battleships. Have not the battle cruisers and submarines proved valuable? While these leaders in Congress undoubtedly place a high estimate upon the importance of the battleship as the primary unit of the fleet, many of them also value our latent resources for the maintenance, repair, and construction of warships, as well

as the facilities for producing the vast quantities of munitions of war required to keep fighting ships upon the battle line.

Has it not been conclusively shown that those naval powers which can not build battleships can not maintain and operate such vessels? Inefficiency and deterioration of all seagoing craft is almost directly proportionate to the knowledge that the respective personnel have of ship equipment and construction as well as to the character of inspection conducted in the repair and building of the vessels. The patient, continuous, and resourceful work involved in preparing the fighting ships for battle and the reserve capabilities of our commercial shipbuilding plants and naval stations to expeditiously refit the disabled ships must likewise be accorded considerable weight in determining the measure of relative naval strength.

Too many experts have a tendency to think too strongly in terms of battleships when measuring naval strength. The fact that strength in battleships is not the only measure of relative naval power was emphasized both in the Spanish-American and the Russo-Japanese Wars. When hostilities began in these conflicts the opposing forces were regarded by naval experts as about equal; yet within a year from the declaration of each war the substantial destruction of the Spanish and Russian fleets had been accomplished.

Courage, patriotism, and knowledge of strategy are fairly general, and therefore the destruction of the Spanish and Russian fleets can not be associated either with the individual heroism or the superior skill of the victors. In these wars, the defeat of the vanquished was not only due to the lack of preparedness, but also to the lack of knowledge of the ships and armaments. Instead, therefore, of dwelling upon relative naval strength solely in terms of warships, it should be measured also from the standpoint of efficiency, industrial resources, and the knowledge of ships and equipment upon the part of the respective personnels.

The demand for more superdreadnaughts has almost be-

come an obsession upon the part of the most aggressive of the advocates of the battleship, and the intolerant manner in which nearly every other suggestion for promoting naval preparedness has been received by these experts is being strongly resented by some of the best friends of the Navy, since the speedier armored vessels of the battle-cruiser type have done about all the sea fighting that has yet taken place in this war. The battleship is a national need, and yet it has marked limitations in providing for national defense.

CAN EFFICIENCY OF OUR NAVY YARDS BE GAUGED BY THOSE WHO HAVE NOT HAD VARIED OR IMPORTANT DUTY AT OUR NAVAL STATIONS?

In general, those who have had but limited experience in the administration of navy yards are the keenest to condemn their operation and organization. Is it surprising, therefore, that we are told that much that we have done in the development of our naval stations conflicts with certain strategic and economic principles? The academic and strategic principles essential to efficient navy-yard organization apply so generally to all industrial operations that they are virtually truisms and platitudes. No one denies assertions of the following nature:

> "That the shore establishments should be made subordinate to the fleet; that every dollar expended for naval purposes should be to obtain the best results; that money should be wisely appropriated and carefully expended; that every dollar wasted on shore is so much deducted from the increase of the fleet; that efficiency and economy will flow from good organization and administration; and that diffusion of effort and unnecessary duplication and extension of navy-yard plants are wasteful?"

Wherein can efficiency or development of naval station or of the fleet be promoted by the utterance of platitudes of such character.

It has been likewise affirmed that in order to locate and develop our navy yards efficiently, there is involved an extensive knowledge of history and strategy, and that it is essential that a careful and exhaustive study should be made of our own national policies and the aspirations of other nations whose interests run parallel to our own purposes and policies.

How about the interests and policy of nations whose purposes do not run parallel to our own? All these things, we are told, must be considered in the solution of the problem of the navy yard. Is it not probably a fact that all necessary industrial and military considerations were given due thought in the location of our various naval stations?

The problem of the navy yard neither involves deep mystery nor profound strategy. It is rather a problem of observation, experience, industrial judgment, and administrative ability.

VIEWS OF THOSE WHO HAVE HAD VARIED INDUSTRIAL EXPERIENCE.

The inexcusable attacks that have been made upon the usefulness of our shore stations can not be based upon knowledge and experience. Should not the views of those who have had important and extended duty in the repairing and construction of ships; who possess knowledge of the demands that were made upon the navy yards during the Spanish-American War; who since then have made a careful study of the problem of navy-yard operations; and who have had considerable sea experience be given an opportunity to be heard upon the subject?

The problem of the navy yard is one of fact; not one of opinion. The industrial demands of the fleet can not be met unless the equipment of the shore stations is extended. It was a grievous military mistake that an extension of the naval shore stations was not effected along with the increase of the fleet. For years, however, the policy has been urged

of increasing the Navy afloat by retarding industrial development ashore. If those who shape our naval policy have not accurate knowledge of the value of our naval stations, then the Congress may inquire if the administration of these navy yards should not be turned over to those who do have a high appreciation of their military value to the fleet. The Navy has much at stake in this matter, for if the existing naval stations are not sufficiently numerous and industrially adequate to meet the needs of a modern fleet, the operations of the fleet will be unsuccessful.

INDUSTRIAL CONDITIONS AT OUR SOUTHERN NAVAL STATIONS A MENACE TO NATIONAL PREPAREDNESS.

It was because Admiral Dewey's squadron had been made "ready" for battle, that Capt. Gridley opened the battle of Manila Bay. This Nation should be "ready" before we invite or accept battle with a formidable foe.

In summarizing the causes which led the French Government to make him "Master of Forges of France," Monsieur Albert Thomas, Under Secretary of State for Artillery and Munitions of War, thus spoke:

> "Today a nation can not consider itself in a state of preparedness for war unless it has planned the mobilization of its industries along the same lines as that of its army and navy. Before the present conflict taught us the meaning of shell power, armies and money sufficed for the conduct of a successful campaign. We still need armies and money, but unless we have behind the trenches an organization capable of producing sufficient armament and ammunition to satisfy the staggering demands of our gunners and riflemen neither troops nor treasury can assure the national defense."

A brief statement of the industrial and military conditions to the southward of Cape Hatteras, with regard to the efficiency of those naval stations, will show surprising con-

ditions concerning naval preparedness. Cape Hatteras constitutes the logical and natural point of division between the North and South Atlantic coasts of the United States. There are about 2,500 miles of sea coast on the South Atlantic and Gulf coasts between Cape Hatteras and the Rio Grande; there are about 1,250 miles between Cape Hatteras and Passamaquoddy Bay, Me. Along the seacoast of the New England and Middle States we have a series of naval hospitals whose average cost will approximate about $300,000 and whose distance from each other only averages about 100 miles, while on the southern stretch of our Atlantic and Gulf coasts the Navy possesses but a series of small dispensaries whose average value has been appraised at less than $10,000 each, and these dispensaries will average about 350 miles apart from each other; below Cape Hatteras we have not a single modern naval hospital. There is not a dry dock possessed by the Government or by anyone else on the South Atlantic or Gulf coasts which will dock any of our superdreadnaughts. There is not a single stationary or floating crane possessed by the Government, by any individual, or by any corporation on the South Atlantic or Gulf coasts which will remove from or install in a battleship either a modern turret gun, a Scotch marine boiler, or an assembled low-pressure turbine of the type now installed either in one of our large naval colliers, tankers, or battleships. For a distance, therefore, of 2,500 miles along our southern seacoast we can not make extensive repairs to any large collier, tanker, or battleship. We have not a single modern naval magazine along this stretch of coast.

The storehouse facilities of the southern naval stations are fatally weak. There is substantially but one naval foundry to the southward of Cape Hatteras, and that is the one at Charleston, S. C., naval station; a foundry whose welfare and sanitary conditions are very bad. Although the Charleston, S. C., Navy Yard is the home port of 12 torpedo-boat destroyers, that naval station has no floating derrick to handle various appliances, such as turbines, evaporators, boilers,

pumps, steering engines, etc., that have to be repeatedly removed and replaced in overhauling destroyers. There is not a single large modern gap lathe, slotting machine, planer, milling machine, hydraulic riveter, hydraulic flanging press, or, in fact, any large power tool installed, at any southern navy yard, capable of doing all the work required upon a modern collier, tanker, or battleship. The inadequate industrial equipment of the southern navy yard may be summarized in the statement that it is not possible, at any naval station south of Cape Hatteras, to build even the cylindrical boilers of a modern seagoing tug.

Is not our naval preparedness a travesty so long as we have a stretch of 2,500 miles of our seacoast provided only with a semblance of naval stations? With an appropriation of $175,000 there can be provided at Charleston, between the navy yard and the sea, a channel 600 feet wide in the straight reaches and 1,000 feet wide at the bends, with a controlling depth of 30 feet at mean low water for the entire distance between the navy yard and the lower harbor. Between the lower harbor and the sea the controlling depth is 28 feet at mean low water. As the rise and fall of the tide is about 5 feet, steamers drawing 29 feet have actually entered the lower harbor and proceeded to their docks for unloading.

The development of efficient naval stations to the southward of Cape Hatteras has been checked and thwarted by the promise to provide a great station at Guantanamo. As substantially nothing of importance has been done as regards naval development at Guantanamo the conditions at the Charleston, Key West, Pensacola, and New Orleans naval stations are simply amazing. Is it not incongruous that naval strategists are constantly affirming that our fleet may engage in great naval conflicts in the Caribbean, and yet no effort is being made to provide a modern dry dock, necessary cranes, and adequate equipment for the naval stations located to the southward of Cape Hatteras?

Should there not be immediately undertaken the extensive

development of the Charleston, S. C., Navy Yard and of at least one of the naval stations on the Gulf? These stations should be provided with extensive deep-water piers, suitable power tools, crane and transportation facilities, modern storehouses, and hospitals, together with dry docks capable of receiving the largest ships that could enter their harbors. A channel at least 32 feet deep should be provided for every harbor where we have a navy yard.

THE STRATEGY OF THE PAST MUST NOT DETERMINE THE NUMBER AND CHARACTER OF OUR NAVAL STATIONS AT THIS TIME.

While the principles of strategy are changeless, their application varies with the military conditions of every age. Our existing and prospective international relations make it impracticable for the limited number of naval stations that have met the needs of a previous decade to suffice for our requirements of the early future. The strategic problem has so many elements that solutions can not often be unreservedly accepted. The Congress, because of such numerous and divergent views of the strategic importance of various harbors in the West Indies, is well aware that the strategic possibilities of particular localities often vanish when viewed from conditions of a following generation or even of a decade.

As evidence of the change of front that has been shown by our own strategists in regard to the military importance of certain harbors in the Caribbean, it is well to recall the fact that during the past 50 years the Congress of the United States has received official communications from the Navy Department concerning the strategical importance of Samana Bay, *Santo Domingo;* Mole St. Nicolas and Cape Haiten, *Haiti;* Santiago, Habana, Guantanamo, and Golf de Guacanabo, *Cuba;* St. Thomas; Culebra and San Juan, *Porto Rico;* Kingston, *Jamaica;* Hamilton, *Bermudas;* and also other points in the Antilles. The sudden change of

front upon the part of some experts as regards the strategic importance of various ports in the West Indies affords sufficient evidence to cause thoughtful men to hesitate in giving weight to purely stategical recommendations.

While representatives of the naval committees may patiently listen to suggestions that would locate all of our Atlantic coast naval bases at points north of Cape Hatteras. it is quite certain that the great majority of both naval committees view with surprise the attitude of naval experts upon this subject. This belief is due to the fact that although there have been numerous attempts during the last 50 years to abandon certain naval stations on the Atlantic coast, the only substantial success attained in this direction has been in the case of New London, and even it was reopened several months ago as a repair base for the submarine flotilla.

Our need of shore establishments has no counterpart in any other navy. As we have over 2,000 miles of seacoast on the Atlantic, over 1,200 on the Gulf, and over 1,000 miles on the Pacific to defend, we need more rather than fewer navy yards. The establishment of a great naval base on the southern California coast is a national necessity, for the distance between San Francisco and Panama is over 3,000 miles. If Germany could afford to spend over $75,000,000 in the upbuilding of Wilhelmshaven, surely we ought afford to spend one-fourth of that amount for the upbuilding of a naval station at some point nearer to Panama than San Francisco.

AT LEAST FIVE GREAT NAVAL STATIONS REQUIRED ON THE ATLANTIC COAST, ONE ON THE GULF, AND THREE ON THE PACIFIC.

At least five great naval stations are required on the Atlantic coast, one in the Gulf of Mexico, and three in the Pacific.

Modern dockyards, both to the northward and southward of Cape Hatteras, must be provided to meet the demands

of any fleet called upon to undertake the defense of the Atlantic coast. If there is not a possibility of our engaging in war with a combination of powers, or with the greatest of naval powers, is not our existing fleet ample for our future needs? The necessity for more battleships is urgent, but even an enlarged fleet is likely to meet with defeat if there is not effected a corresponding development of our naval stations.

As regards the Boston and New York Navy Yards, due to their comparatively limited areas, efficient extension can only be accomplished to a limited degree.

When reviewing the navy-yard problem as a whole, it is well to remember that the terminal commercial water frontage and pier facilities at practically all of our leading seaports on the Atlantic coast are becoming even now inadequate to handle expeditiously the increase of freight traffic at these ports. The collector of the port of New York has stated that the commerce of that city is already congested; and that some of the commerce of the port should be deflected to other harbors. It has been asserted by transportation experts that unless the wharf terminal transportation facilities of some of our eastern cities are soon extended, it may be only a few years before water-front traffic congestion may seriously affect the cost and distribution of the food supply of those ports. It is, therefore, possible that, even within a single generation, the demand may come from the maritime associations and the railroad transportation systems of the country that the question of abandoning certain navy yards should be given serious consideration by the Navy Department, since the extension of one or more miles of commercial quay-wall facilities may have a far-reaching effect in improving the transportation facilities of ports like Boston and New York and in lowering the cost of living.

If our naval stations are to be developed only to meet certain artificial and favorable conditions conforming to a regular docking schedule, whereby the various ships of the

fleet can be sent to particular navy yards for repairs at regular periods of each year, it is certain that these stations will be found woefully deficient in times of emergency. Under the stress of war, regular docking schedules cannot be maintained. Both the character and extent of the repairs required upon battleships in war time will be much more important than when the ships are operated in peace.

Fleets and naval stations exist for the needs of battle. Hence the military and industrial resources of the navy yards must be adequate to satisfy emergency war requirements, and a large reserve of shop equipment must be provided for each naval station. In order to prepare the naval stations for such conditions the initial outlay for development may seem excessive as compared with the limited expenditure that might be required to keep them going to meet peace requirements.

OUR LACK OF MODERN DRY DOCKS.

The United States now possesses but three dry docks capable of receiving the largest battleships under construction. These three graving docks, located at New York, Norfolk, and Puget Sound, will not meet our naval needs even of a few years hence. A fourth dock has been authorized for Pearl Harbor, Hawaii, but it is not due for completion until July 1, 1918. Only one of these docks is over 828 feet long and therefore may be the only one that will dock certain war vessels that we are projecting.

Probably one-half of the coastwise ship construction of the Atlantic coast is done on the Delaware River, and yet neither the Philadelphia Navy Yard nor any private shipbuilding plant on the Delaware River possesses a dry dock capable of receiving our latest battleships.

Germany has docks at Hamburg, Kiel, Bremerhaven, and Wilhelmshaven. With one-twentieth of the seacoast that we possess, Germany has today one more large dry dock than the United States.

We have been able, after a fashion, to dock all our ships

by maintaining a special docking schedule whereby the respective ships of each squadron can be sent in turn to particular navy yards at regular periods of each year. Independent of the additional large modern graving dock that is projected to be built at Norfolk, does it not appear imperative that equally large dry docks should likewise be provided at other points on the Atlantic and Gulf coasts and particularly at Philadelphia, Charleston, and at some point on the Gulf? At least two more such docks are needed in the Pacific. If each annual naval appropriation bill for the next six years contained an authorization for a new dry dock, an interval of at least 10 years would elapse before the last of these docks could be completed.

The industrial and military importance of constructing a great naval base at Pearl Harbor cannot be overstated. We can never hope to be any important factor in controlling the commerce of the Pacific until we provide assured docking facilities at Hawaii, and for a time at least the docking of all very large merchant vessels at American ports on the Pacific will have to be done at our navy yards.

If we are to have a Navy capable of protecting our outlying naval stations and colonial possessions, it is as imperative to adequately develop naval bases as it is to build battleships. It is simply recognizing the inevitable, however, to appreciate the fact that the complete industrial and military development of the naval stations on the South Atlantic, Gulf, and Pacific will not be sacrificed to build up Guam and Guantanamo. Such action would not commend itself to the spirit and purpose of America.

As regards the graving docks at Panama, these docks would probably, in time of peace, give preference to the merchant vessels passing through the canal. The possibility of the destruction of the Isthmian dry docks, taking into consideration their location and relation to the canal itself, should make us somewhat apprehensive as to whether these docks will, in time of war, reliably serve all the far-reaching purpose that prompted their construction.

The naval dry-dock problem is exceedingly important from an industrial, military, or strategic standpoint. It can also be expected that there will be periods when one or more of the modern naval dry docks will, from some unavoidable cause, be found temporarily unserviceable. The dock repairs upon a single vessel might extend over a period of many months.

The question is thus well worthy of consideration as to whether both naval and national interests would not be subserved, by deflecting, if necessary, the appropriation for at least one battleship to the construction of four or five modern dry docks. It is well to remember, also, that the time required to build a modern dry dock will exceed that required to build the largest battleship contemplated. Further, it is probable that, under special contingencies, we might purchase battleships abroad in advance of a formal declaration of war. We sold, under approximately these conditions, the battleships *Idaho* and *Mississippi*.

If various Commonwealths, municipalities, or commercial interests, however, can be induced to build such docks, under special agreements with the Navy Department, the number of naval docks that should be built by the Government in those special localities might be correspondingly reduced. All new graving docks should be capable of receiving the largest steamers that we might hire or purchase for naval or military purposes.

As the commerce of the South Atlantic and Gulf coasts does not, and may not for years, justify the construction of modern commercial dry docks in those locations, the building of such structures at our naval stations on the South Atlantic and Gulf coasts should be promptly begun. The restoration of the American merchant marine upon the high seas would be greatly promoted if all our naval stations possessed modern dry docks, deep-water piers, and powerful stationary cranes which could be utilized by the commerce of the respective ports.

With an increase in the number of dry docks, there should

be provided more piers at all our naval stations. The naval demand is but following the extensive commercial demand for more and deeper piers. The increased pier facilities being made at every great port of the world is index of what we need at our naval stations.

DOCKYARDS POSSESSED BY GREAT BRITAIN.

We may well take a lesson from England with regard to the industrial and military value of the dockyards. When that nation regarded France alone as its probable naval foe, the Admiralty established dockyards at Chatham, Haulbowline, Pembroke, Keyham, Portsmouth, and Sheerness. With the revival of the United States as a naval power, England commenced the development of dockyards at Halifax, Bermuda, and Esquimault, and undoubtedly had expectations of establishing such a base at Hawaii.

For naval and maritime purposes in the Far East, England established a naval station at Hongkong. For the future protection of South Africa and Australia, there was built the extensive station at Simons Bay, Cape of Good Hope. For naval operations in the Mediterranean, there were developed the dockyards at Gibraltar and Malta.

With the advent of Germany as a possible naval rival, the British Admiralty commenced constructing on the east coast of Scotland the great naval station of Rosyth, whose development up to 1914 contemplated an expenditure of $24,000,000. Four graving docks, capable of receiving any British vessel, afloat or projected, are now under construction, or have been completed at this station.

Even before the commencement of this war there had been expended about $10,000,000 upon the dockyard at Simons Bay, Cape of Good Hope. At Portsmouth, England, two new dry docks have been lately completed at a cost of $6,500,000.

England has likewise hundreds of commercial graving and floating docks, and thousands of berthing piers for

her mercantile marine, which structures could in part be used for naval purposes. The Admiralty, however, deemed it necessary to develop naval dockyards both at home and throughout the world, although there was available for the use of the British fleet the right of way to the many colonial shipbuilding plants controlled by her subjects.

In contrast to our apparent indifference to dry-dock construction Great Britain has systematically carried on the work of dry-dock extension along with the augmentation of her fleet. Never has the Admiralty outlined an extensive program of warship construction without providing for additional dry docks to receive the battleships authorized.

At no period, however, in her history has Great Britain pursued any policy that would encourage the development of outlying naval bases at the expense of the home stations. Her policy has been to first make her home coast both an impregnable wall of defense and a base for shipbuilding, and then to take up the question of the development of the outlying naval bases. Dominance of distant seas has followed, not preceded, the defense of the home coast.

WHAT CONSTITUTES A COMPLETE MODERN NAVAL STATION?

If most of our various naval stations were really what they were presumed to be, it might appear that we had a considerable number of navy yards. Considered, however, from the standpoint of lack of dry-dock and berthing-pier facilities, or from the inadequate character of the industrial shops for meeting war requirements, then the extent of our naval industrial activities is decidedly more limited and unsatisfactory than is realized.

It is not possible to develop, in a brief period, any great naval dockyard or naval base anywhere in this country. England's experience in building a great dockyard at Rosyth in Scotland is illustrative of the extended time required for completing such a project.

The Board of Inspection for Shore Stations gave special

consideration to the question of the requirements demanded of a great naval base. Upon this particular subject the Board reported as follows:

> "It would require an expenditure of probably tens of millions of dollars to develop any great station in a manner whereby it might contain nearly all the military and industrial advantages now possessed by even a majority of the separate navy yards. No single station that has been established, or even contemplated, could possibly possess even a majority of the following advantageous features now obtainable in part at some existing navy yard: Security from bombardment; sufficient depth of water adjacent and in approaches, combined with ample maneuvering and anchorage area for a large number of vessels of deep draught in or out of trim; sufficient water front with satisfactory contour for berthing a large number of vessels of deep draught adjacent to the industrial shops; sufficient territorial area for the necessary dry docks and workshops, storehouses, quarters for officers and men, and drill grounds; good foundations for dry docks; docking at any stage of the tide; good transportation facilities, both by rail and water; good material market near at hand; a large skilled-labor market; good, healthful climate, with little interference to out-of-door work; a plentiful supply of fresh water. No nation possesses or probably ever will possess such a naval station, and it may be that all-around naval efficiency is, under certain contingencies, as well promoted by having these advantageous features distributed at various stations somewhat remote from each other as would be the case if all were concentrated at one place."

The development of a great naval base is a project that is very difficult to accomplish, particularly for the United States, since we have comparatively few great harbors. Neither the anchorage grounds of New York Harbor nor

Narragansett Bay, considering the extensive demands of commerce and the channel conditions at those ports, warrant the consideration of either place as a possible great naval base. Has not the commander-in-chief of the Atlantic Fleet already spoken of the lack of anchorage grounds in Narragansett Bay?

CONCENTRATION OF NAVAL STATIONS.

Every maritime nation is eventually compelled to aspire to the possession of an efficient navy, whose development necessitates a system of naval bases.

The considerations governing the location and development of primary and secondary naval stations are so varied, numerous, and important that there always will be a difference of opinion as to where they should be located. Certain of our naval experts formulated the conclusion about three years ago, that only two are required on the Atlantic coast. This conclusion will never commend itself either to the Naval Committees of the Congress or to the people of the United States.

The country will not depend for its production of munitions of war and the repair and construction of ships on a narrow stretch of land between Cape Cod and Cape Henry.

We might have a great army and a great navy, and still be at the mercy of an enemy, unless there has been adequately developed at other points on the Atlantic coast and the Gulf of Mexico, as well as on the Pacific coast, efficient naval bases.

TWO GREAT HOME BASES CAN NOT POSSIBLY MEET AMERICAN NAVAL DEMANDS ON THE ATLANTIC COAST.

Within a year from the time when Narragansett Bay and Norfolk were recommended as the logical sites for the two great home bases contemplated, our experts had changed front on this question by substituting New York for Narragansett Bay. When it is remembered that the Capes of

Virginia are less than 300 miles distant from Montauk Point, the country will never accept the conclusion, that two great home bases so close to each other as either New York and Norfolk, or Narragansett Bay and Norfolk, could meet the industrial demands of a fleet large enough to protect our interests on the Atlantic coast, at the Canal Zone, and in the Caribbean.

The suggestion to concentrate at two great naval bases the important industrial facilities necessary for the Atlantic fleet will never develop into a verity. Such a suggestion will never commend itself to a national body like the Congress of the United States. Our navy yards should not, can not, and never will be concentrated by that body in any limited regional district of a few hundred miles, for such a suggestion will not stand the test of either study or time.

As regards naval operations on the Atlantic coast, our naval policy of defense must be based upon several possibilities. The first possibility, although it may be remote, is that we may have to contend on the sea with a nation operating both from the North and the South. For the first contingency there should be provided a strong naval base at some point north of Nantucket and therefore the Portsmouth, N. H., station should be extensively developed. For the second contingency, since every recognized naval strategist regards the Caribbean as the probable center of great future naval operations, and since we must always keep a strong fleet available for the defense of the Isthmian Canal, one great naval base should be established on the Atlantic coast somewhere between Cape Hatteras and St. Augustine and another on the Gulf of Mexico.

The nation at large will never commend a policy that relies upon two home bases, located between the Virginia Capes and Nantucket; for the defense of the Isthmian Canal; for naval operations in the Caribbean; for the protection of our possessions in the Antilles; and for the defense of the Atlantic and Gulf coasts. The proposition simply discredits

the judment of naval officers in general. In expressing this view, one need not depreciate in any degree the great military possibilities of the lower Chesapeake Bay as a concentration and maneuvering point for the fleet.

When there is considered the extent of our coast line on the Atlantic and Gulf coasts; when it is remembered that certain seaports of our neighbors and important harbors in the Carribean and South America might be used for bases against our fleet, this country will never rest content to rely for its defense along the whole Atlantic, upon two naval home bases, both located north of Hatteras.

LIMITS OF THE ANCHORAGE OF A NAVAL STATION.

One of the reasons advanced for recommending New York Harbor, as a site for one of the two great home bases, was that the anchorage of this base would extend from the fortifications at Sandy Hook, through Long Island Sound, to the fortifications of the eastern approaches on Fishers, Plum, and Gull Islands. If such is the extent of the anchorage of New York, why do not our battleships, when proceeding from the New England ports to New York, make their way to the latter navy yard through the short and direct passage through Long Island Sound? Why does not the anchorage of Norfolk extend both northward to Baltimore and eastward to Lynn Haven Bay? Why does not that of Philadelphia extend to the capes of the Delaware; that of New Orleans to the Passes, and northward to Baton Rouge; and that of Key West to Miami? Does not the anchorage ground of Mare Island, according to this reasoning, extend to the Golden Gate; that of Narragansett Bay to Block Island, Nantucket, and Providence; that of Charleston, S. C., far up the Cooper River; and that of Bremerton to the entrance to Puget Sound? If such extreme liberality is extended in determining the anchorage grounds of the New York Navy Yard, why should not equal consideration be given to extending the anchorage facilities

of every other naval station that we possess? As a matter of fact, the anchorage of a naval base only extends to a point where the resources of the harbor can be efficiently utilized in provisioning, coaling, and handling the supplies and repairs of the ships.

IS IT ECONOMICAL OR DESIRABLE TO CONCENTRATE NAVAL STATIONS?

The doctrine which was preached about two years ago, that the fewer the naval bases the more economical their operations, has about been exploded. Is there one European nation now at war which would not have regarded it as of great advantage to have more widely dispersed their manufacture of munitions of war? The capture of the Belgian and Serbian arsenals (for in each of those countries the ordnance factories were concentrated in a single city) affords all the evidence necessary to dispel the fiction that concentration of military industrial activities is compatible with national safety. There is not a large industrial manufacturing concern in this country which does not find it even of financial advantage to distribute its industrial activities to points far distant from each other. The Locomotive Trust now builds locomotives in New Hampshire, Massachusetts, New York, Pennsylvania, Ohio, and other States east of the Mississippi. Why does not this corporation concentrate its activities in a few places?

THE NAVAL STATION AT PORTSMOUTH, N. H.

Of all our navy yards, the one which has been subjected to the most continuous undeserved attack is that at Portsmouth, N. H. It is pictured as a naval station whose channel is not only tortuous and dangerous, but one which freezes up in winter, while as a matter of fact, due to the special tidal currents of the Piscataqua River, this harbor has only frozen up once during the past 50 years.

The channel leading to this navy yard from the sea is

500 feet wide—wider than the channel leading to either the Boston, New York, Philadelphia, or Norfolk Navy Yards. The controlling depth of the Portsmouth Channel, at mean low water, is 40 feet. It is one of the easiest channels on the North Atlantic coast to enter and depart from. Is not Gibraltar nearer the sea than Portsmouth, and are not the buildings and docks of Gibraltar exposed to gunfire from seaward? Despite the criticism of those who apparently have no knowledge of its industrial possibilities, the Portsmouth Navy Yard will yet develop into one of the most important that we have. By developing the Isles of Shoals into a modern destroyer and submarine base, no enemy would even dare to attempt the blockade or bombardment of Portsmouth. The marked manner in which the British fleet steers clear of Heligoland and Wilhelmshaven best tells of the military possibilities of the Isles of Shoals in protecting the New England seaports to the north of Cape Cod.

One of the most important means of providing for the defense of Portland and Boston would be to develop, at the Isles of Shoals, a great base for submarines and destroyers. With the resources of the Portsmouth (N. H.) Navy Yard to sustain such a base it would be a reckless commander-in-chief of an opposing fleet who would attempt a blockade or attack upon any part of our coast within 50 miles of this offshore base.

The naval station at Portsmouth, however, is there to stay, since it is urgently needed to meet the naval demands of the future. This navy yard needs another dry dock, together with additional berthing piers and industrial shops. The aspersions cast upon the military value of this naval station tells of the lack of knowledge of those who have attempted to check its development. At one-third the cost of a battleship there could be constructed at this navy yard a modern dry dock, likewise three deep-water piers, necessary industrial shops, together with the development of the Isles of Shoals as a base for submarines and destroyers. The

program for the enlargement of the fleet ought not to be reduced, but the writer believes that the naval shore development and extension thus recommended would mean more to national defense than the building of a battleship.

NAVY YARD, PHILADELPHIA.

The extensive water front of this station, its proximity to many of the leading commercial shipyards of the country, its advantage as regards obtaining material and labor, its immunity from capture, its extended territorial area, and its general possibilities of development make it the logical navy yard for the construction of battleships and other large types of naval craft of heavy displacement. It possesses the most distinct advantages for the location of the large central foundry that is an imperative naval need. It should be developed likewise into the most important manufacturing naval station that we possess. It is lamentably weak as regards its industrial shops, the design of these buildings not being conformable to the engineering needs of today. It is inexplicable that this station does not possess a modern stationary or floating crane of high hoist, long reach, and of 200 tons capacity. Even more inexplicable is the fact that this station does not possess a modern dry dock, capable at least of receiving the largest dreadnaughts now under construction for the Navy.

NAVY YARD, NORFOLK.

The inestimable advantage and superiority of Hampton Roads as an assembling and concentration base for the fleet make it certain that this navy yard should be developed into the most important one that we possess on the Atlantic coast, particularly as regards the feature of "supply," which embraces emergency repairs and docking of hundreds of vessels. It must be a source of wonderment to shipbuilding and engineering experts to behold a collection of antiquated and even unsanitary industrial shops absolutely unsuited for

the important work conducted at the yard. The quay-wall facilities should be materially extended, since this station, of all others, should have a reserve of berthing space to meet the unexpected and excessive demands constantly arising in this direction. The industrial needs of this navy yard were specifically pointed out in a report submitted in 1914 by the Board of Inspection for Shore Stations. Its transportation demands are of the most extended character, justifying along this line heavy expenditure.

OUR SOUTHERN NAVAL STATIONS.

For about 10 years some of our strategists were calling for the permanent closing or abridgment of the industrial activities of the southern naval stations. Fortunately for the interests of the Navy and the Nation, the effort to close the Charleston (S. C.) Navy Yard was thwarted through the far-seeing judgment and determined opposition of Senator Tillman. With regard to the naval stations at Key West, Pensacola, and New Orleans, the operations of all three stations should be materially extended.

Either at Pensacola or at New Orleans there should be developed a large naval base to meet the vast military demands that will eventually arise in connection with the defense of the Isthmian Canal. Whether or not we need an outlying naval base at Guantanamo, the protection of the Isthmian Canal demands the development of a large naval base on the Gulf of Mexico.

One of the ablest American writers and authorities upon naval affairs is Prof. William L. Cathcart, a graduate of the Naval Academy, and a former Officer of the Navy. In writing upon the Panama Canal as a breeder of war, Prof. Cathcart thus comments:

> "With regard to the future of the canal there are two facts which cannot be emphasized too strongly. First, barring the Monroe doctrine, there is no policy or possession of the United States which is more likely

to breed war for its retention than the world prize—the canal Second, should we be too weak to defend it in war, all of its military advantages would pass to the enemy which took it. In that event its construction will simply have weakened us immeasurably, for through it we shall have opened a gateway for the nations of Europe to our Pacific coast and our overseas possessions beyond.

"That these forecasts have sound bases in history the story of the Suez Canal proves. It was built for Egypt by France; it has passed, seemingly for all time, to England. The vital necessity for a British highway to India predetermined its ultimate ownership. Beginning with Gibraltar, England slowly through long years acquired Malta, Cyprus, Perim, and Aden. The missing link was the canal, and Disraeli secretly bought a large, almost a controlling, interest in its shares. Then, in 1882, England entered Egypt 'to restore order.' Now, in 1916, after uninterrupted occupation, she proclaims a protectorate over that land of sun and sand, and holds the canal in a grip that will never loosen.

"For fifty years and more the history of the Caribbean has been marked by European intrigue. Were it not for the European war we should now have active demonstration of this influence in Haiti and Mexico especially. With this world problem looming large in the near future, with that world prize—the Panama Canal—as the chief stake, is it likely that a country in a military and naval sense so weak as ours will be allowed to hold for long its present predominance or its coveted possessions in this sunlit sea?"

The Canal Zone can only be defended on the Atlantic side from a naval base located on our own shore. It can, and possibly will, be attacked from the Pacific coast side. As nearly all the industrial establishments of this country

are east of the Missouri and Mississippi Rivers, the logical naval base for the defense of the Atlantic end of the canal is at some coastal point on the Gulf of Mexico.

THE DEVELOPMENT OF THE CHARLESTON, S. C., NAVAL STATION A MILITARY NECESSITY.

Charleston, S. C., has the same tariff freight rates to the coal fields of Virginia as Norfolk. As it has likewise modern coaling piers, high-grade fuel can be shipped as cheaply from Charleston as from Norfolk or Newport News. It is an extensive and important center for distributing fuel oil.

Every railroad system traversing the coastal region between the southern Appalachians and the sea has terminal piers at this port. It has more deep-water frontage and deep-water anchorage ground than any harbor on our South Atlantic coast. It is the logical and national deep-water Atlantic terminal seaport for the industrial activities of East Tennessee, South Carolina, and the southern portion of North Carolina.

No American fleet could operate for a protracted period in the Caribbean without establishing on the Atlantic coast a naval base south of Cape Hatteras, and Charleston is the only port on that stretch of coast which can be adequately developed into such a base.

Despite the expenditure involved in such development (and the ultimate outlay should not be less than $8,000,000), this important work should be commenced without delay. A modern dry dock should be constructed; at least two deep-draught battleship piers built; a 150-ton stationary or revolving crane installed; additional industrial shops and storehouses obtained; a magazine for powder and projectiles provided; a suitable hospital established; additional barracks erected; and the channel to the sea improved. The development of this station means more to naval preparedness than the building of even two battleships, for it

would compel every possible naval foe to extend its battle line several hundred miles, and at least a half dozen more opposing ships would have to be employed in such service.

THE STRATEGIC IMPORTANCE OF DEVELOPING THE KEY WEST NAVAL STATION.

Probably no naval officer has given more intelligent and thoughtful consideration to the study of the military and strategic importance of Key West as an important and necessary naval base than the late Commodore William H. Beehler, United States Navy. His reports are worthy of careful consideration.

It is regrettable that in a general review of the navy-yard problem only brief reference can be made to the exhaustive, thoughtful, and valuable study of Commodore Beehler as to the strategic importance of this harbor.

The development of the Key West Naval Station is in some respects a distinct military problem. Its claims should also be carefully considered, however, in connection with the extensive work which should be immediately commenced at Charleston, Pensacola, and New Orleans.

The report of the Board of Inspection for Shore Stations dated August 9, 1913, shows the necessity of developing this naval station. At least $2,000,000 should be expended at Key West for piers, berthing slips, and industrial shops, together with the building of a breakwater and basin for the safer berthing of destroyers, submarines, and other Government vessels. In some respects one of the most important demands of the Navy is the immediate development of Fleming Key as an assured and extensive reserve supply of fuel oil, coal, and ammunition. There is no feature connected with the development of our shore stations that should be given precedence to the important work that should be undertaken at Fleming Key, since the enlargement of the fuel base at Key West may vitally concern the operation of the fleet in the West Indies.

The West Channel entrance to Key West, when properly marked by buoys, is good for 28 feet. With a moderate amount of dredging, followed by dragging, an assured depth of 30 feet can be obtained. The importance of commanding the Florida Channel has always been recognized by our naval experts, and Key West should be made the primary home base of a very large number of destroyers and submarines.

NAVAL STATIONS AT PENSACOLA AND NEW ORLEANS.

The military necessity for closing the Pensacola and New Orleans naval stations was thus urged by one of our naval experts less than two years ago, who wrote:

> "To consider any point so far withdrawn from the inevitable field of action, in case of a naval war in the Atlantic, as are either Pensacola or New Orleans as a base, is not only to violate every principle of reason but of common sense as well. That neither is needed for the maintenance and upkeep of the fleet has already been demonstrated by their actual closure for several years. The only logical conclusion that can be drawn from this is that economy of means and the best good of the nation require that their closure shall be permanent and that both should only be viewed as fortified ports of refuge in case of general disaster."

As pertinent to the above opinion, which has been absolutely discredited by the naval events of the past year, it may be well to inquire if operations off the east Mexican coast may not not be regarded as naval operations in the Atlantic. The magazine containing such expression of these views had probably not reached some of its distant subscribers before the Congress, upon the recommendation of the Navy Department, and under the stress of possible temporary occupation of certain Mexican seaports, reappropriated all unexpended balances for the development of the New Orleans station, which had either been unused or had lapsed.

If we require the services of gunboats, torpedo boats, destroyers, colliers, repair ships, hospital ships, and transports for service in the Gulf and in the Caribbean, is it not a military necessity that the splendid floating dock at New Orleans should be utilized to its maximum capacity, and that the industrial shops of that station should be reopened and enlarged in order to make necessary repairs to the vessels that will have to be docked at that station?

Has not likewise the naval station at Pensacola within the past year been reopened as an aviation center, and is there not already urgent call for an increase in the facilities for aviation experimental and research work? Would any naval officer now call for the permanent closing of the Pensacola Navy Yard, and would not an outlay of at least several million dollars at that station for aviation and other purposes promote national preparedness?

OUR NAVAL STATIONS ON THE PACIFIC COAST.

For every important reason that can be advanced for the development of the eastern navy yards, some equally convincing reason can be given for the improvement and extension of the Pacific naval stations.

In view of the slides that have occurred at the Canal and the dynamiting of munition factories in the United States during the past year it will be seen that the Panama Canal may be so seriously obstructed or injured as to entirely prohibit traffic passing through it at a critical time.

Measured, therefore, from different, although equally important viewpoints, the problem of the navy yard on the Pacific coast is as critical and far-reaching as that on the Atlantic and Gulf. Could not, therefore, an expenditure approaching that required for the construction of a battleship be profitably applied in the more complete development of the Puget Sound, Mare Island, and Pearl Harbor naval stations as well as in the establishment of one to the southward of San Francisco?

COMPETITION SHOULD BE DEVELOPED BETWEEN THE NAVY YARDS AND THE COMMERCIAL SHIPBUILDING ESTABLISHMENTS.

The future interests of the commercial shipbuilding establishments engaged in naval construction, as well as the efficient development of the navy yards, will best be promoted if there is well recognized and fair competition between the navy yards themselves, as well as between the navy yards and the private shipyards.

While it is admitted that certain commercial establishments, as well as particular navy yards, by reason of local and other conditions, can do work along certain lines, in a more efficient and economical manner than can be done elsewhere, no such establishment, whether commercial or national, should be given a monopoly of any form of ship construction or industrial manufacture. There should be certain well-established principles enunciated by the Navy Department, in regard to the distribution of naval work, and amongst these principles should be the following:

(*a*) In general, the fact should be recognized that the navy yards can not turn out work more expeditiously and economically than the commercial plants. The problem of what constitutes indirect and overhead charges is viewed so differently by the naval officials and shipbuilders that competitive bids received from navy yards and commercial shipyards can not tell the actual cost of work done at the respective plants.

(*b*) Under no consideration should there be assigned to the navy yards more than one-half of the work involved in the building of any type of fighting craft or naval auxiliary, or in the manufacture of naval guns, war munitions, or industrial appliances of any character whatever.

(*c*) Where the estimated cost of navy-yard work exceeds by 20 per cent. the bid by commercial plants, the navy yards should in general be debarred from being assigned the manufacture of such appliance.

(*d*) Even where any particular navy yard shows its ability to manufacture certain appliances in a more economical manner than at other naval stations or commercial plants, not more than one-half of such work should be assigned that particular naval station. The remaining half of such work should be distributed amongst other naval stations or given to private firms, in order that a healthy competition for the manufacture of such articles might be aroused. Such a distribution of work would also provide for emergency conditions, as in the case of the accidental or wanton destruction of buildings or machinery of any commercial or national establishment. The destruction of any plant having a monopoly or doing the bulk of any kind of work would interfere for a considerable time with the production of an adequate output of some important article of supply required for the needs of the fleet. We ought therefore always provide reserve plants for the manufacture of every article or appliance required by the Navy.

(*e*) The field of competition as regards the manufacture of all naval appliances should be kept open and made as broad as possible. The development of at least two navy yards on the Atlantic coast and one on the Pacific to carry on every distinctive military feature of manufacturing enterprise would always insure to the department an efficient, expeditious, and adequate output of necessary and important supplies.

(*f*) The department should thus establish a definite policy of refusing to definitely assign to any navy yard the exclusive manufacture of any special article. There are, of course, stations like the Naval Gun Factory that were established for a definite industrial purpose, and work of that character should, as a rule, go exclusively to such plants. And yet there are industrial experts who are amazed that an establishment of this character was located in a city that advertises itself as a social and official center.

(*g*) It may be that, through changes in the directive and administrative personnel of certain navy yards by the as-

signment of special work to these places and by an extension of their equipment, certain stations might be developed to a condition whereby they could be enabled to carry on work in a far more satisfactory and expeditious manner than had been done under previous conditions. Every station should clearly be given to understand, however, that it had no vested right to a monopoly of work of any kind, and that the failure to do work in an economical, expeditious, and efficient manner would result in diverting the work to stations where more satisfactory results could be obtained.

(*h*) Even such establishments as the Washington Gun Factory and the torpedo station at Newport should not be given a monopoly of auxiliary work of a character like that of the manufacture of torpedoes, projectiles, shell, powder cases, etc., for there is little doubt that all of such work could be done as expeditiously and as economically at various navy yards as at distinctly ordnance plants, particularly as these special ordnance plants are not located in the vicinity of the metal-industry centers.

NO NAVAL STATION SHOULD BE SPECIALLY FAVORED IN CONTINUITY OF EMPLOYMENT OF ITS ARTISANS.

It is important to note that at certain naval stations, and this has been notably the case at the Washington Gun Factory, discharges in the workmen's force, for the past 12 years, have probably not varied even as much as 5 per cent., while at other naval stations the variation has been as great as 50 per cent. It surely seems important that the skilled technical complement of one naval station should be kept intact as well as that of another. In fact, if any discrimination is to be shown it would appear that the smaller yards might be given the preference in this respect, since skilled mechanics, discharged from the larger navy yards, ought to have a more reasonable chance of securing employment at their own or at assimilated trades, in larger cities, than they would have in the smaller towns.

In order to insure more steady employment at all our navy yards for a complement of skilled artisans, the manufacture of cartridge cases, spoons, torpedo tubes, powder, shell hoists, and other ordnance accesories, now manufactured at the Washington Gun Factory, should be deflected to other naval stations. There are no impelling industrial or military reasons why all such accessories should be made in the same establishment where heavy ordnance is manufactured. There is some work being done daily at every naval station which is as difficult as gun construction, and therefore there is no doubt but that such ordnance accessories could be manufactured as cheaply, expeditiously, and as efficiently elsewhere as at the Washington Gun Factory.

THE EFFICIENT TREND OF NAVY-YARD DEVELOPMENT.

Any destructive criticism of administrative and constructive methods is unwarranted, unless the critic makes some recommendation that promises better results. Those who have had extended experience at navy yards have recommendations to make that are practical in character and are founded upon as much unselfishness and patriotism as are the suggestions made by others whose knowledge of navy-yard affairs is not based upon actual technical training in industrial affairs.

It would be conducive to efficiency to use as a precedent, in formulating some broad basic plan of nayy-yard development, the far-reaching and thoughtful recommendation of Rear Admiral N. E. Mason, United States Navy, when reporting upon the most desirable trend of development of the Washington Gun Factory.

Admiral Mason suggested that the gun factory should be developed and extended to an extent that would permit it to manufacture all gun work then required by the Navy by working double time (16 hours). When this had been accomplished, he recommended that all extensions of the plant should cease, and that the gun factory should then operate

only on an 8-hour basis, thereby permitting the remaining heavy ordnance work required by the Navy to be given to outside commercial firms. And it is along such constructive lines that all our navy yards should be developed, thereby arranging that about half of all our naval construction work should be given to commercial establishments.

FOUNDRY EQUIPMENT OF OUR NAVAL STATIONS ARE ABSOLUTELY INADEQUATE FOR THE NEEDS OF THE FLEET.

Over two years ago the Board of Inspection for Shore Stations submitted a series of reports concerning industrial conditions at each naval station on the Atlantic and Gulf coasts. A general analysis of these reports show that our naval stations had not been concomitantly developed with the augmentation of the fleet.

While the shore board insistently recommended the installation of more heavy power machine tools; the construction of additional berthing slips, building ways, and dry docks; together with the erection of modern storehouses and industrial shops, there has not been authorized by the Congress during the intervening period any of the structures thus recommended, except building-ways for five naval stations.

As regards naval foundry conditions, affairs were found so unsatisfactory and inadequate that the board was constrained to submit a special report upon this particular question. The following is an abstract from this report:

> "There is not a single naval foundry on the Atlantic coast which is capable of producing a casting for a steel stem or sternpost of a battleship. There are probably not over two foundries which are capable of casting a low-pressure cylinder of the size of those installed on some of our battleships and colliers, and then it would require a length of time that would be intolerable under war conditions. In regard to the low-pressure turbine casings of the latest battleships au-

thorized, it has been necessary to divide both the upper and lower casings into sections fitted with longitudinal flanges in order to bring the work within the capabilities of the foundry at the New York Navy Yard.

"Not one single naval foundry, except that at Washington, D. C., possesses a special room for cleaning castings in a sanitary and efficient manner, and there is not a modern sand-mixing room at any station. There is not a station which possesses either a suitable flask yard, or an efficient, economical and expeditious arrangement of concrete bins for the delivery, discharge or storage of bulky and necessary foundry materials. In general, the naval foundry conditions on the Atlantic coast constitute a travesty upon economy and efficiency, and the more carefully the matter is investigated, the more urgent appears the necessity for immediate action.

"The foundry demands that now exist are small as compared with the demands that would come in time of war. The delay resulting, moreover, in producing special castings by resorting to temporary expedients, is not in line with naval efficiency. In time of emergency, such delays might readily constitute a serious bar to naval operations. While the direct cost of producing castings under present conditions may also appear reasonable, as measured by simple cost account methods, the delays frequently incurred, in improving methods and in effecting temporary changes, have probably been of an extent that makes the indirect cost of such castings exceedingly heavy. There is no more reason why delay should be tolerated in the foundry, in resorting to temporary expedient on account of lack of facilities and equipment, than that it should be permitted to exist in the machine shop, since delay upon naval work is very expensive, particularly when it is remembered that the cost of maintenance of a modern

battleship will approximate several thousand dollars a day.

"In brief, the naval foundry situation, in general, is such that it is not at all improbable that one of our battleships might have to wait six months, if not longer, for a steel stem or sternpost, shaft struts and tubes, turret roller tracks, low-pressure cylinder, or for certain turbine casings. The delay experienced in obtaining the naval castings is due to the fact that the Navy requires material that possesses a maximum of strength with a minimum of weight; and this calls for the highest grade of castings, both as to quality and as to freedom from distortion."

Can it be doubted that the military and industrial need for foundries is so urgent that the department would be justified, if funds could not be obtained otherwise, in reducing the projected increase in the number of destroyers from 50 to 47 in order to improve the foundry situation?

The remarkable advance of Germany as an industrial power can in part be ascribed to the expeditious and economical manner in which her foundries can turn out reliable, complicated, and massive gray-iron and steel castings. For nearly a generation in advance, of other industrial nations, Germany had a realization of the fact that it was in the foundry department of modern industrial and shipbuilding plants that the greatest savings could be effected. Does not an exigency exist which would warrant this nation in making some distinctive development as regards the problem of the naval foundry?

INSUFFICIENT APPROPRIATIONS ALLOTTED FOR THE UPKEEP OF OUR NAVAL STATIONS.

The following is an extract from the report of the Board of Inspection for Shore Stations as regards conditions at the navy yard, New York, concerning the repair and preservation of buildings, and the conditions stated are applicable to other navy yards:

"Attention is called to the fact that there has only been allotted for the repair and preservation of the buildings, docks, piers, streets, and other public works utilities of this yard for the fiscal year 1913 about one and one-seventh per cent. of the total value of the plant —an amount absolutely inadequate to properly preserve Government property. The fact is still more startling that the total appropriations for repairs and preservation of all public buildings in our navy yards is only $800,000, or about four-fifths of 1 per cent. of the estimated value of the structures.

"The civil engineer of the station estimates that the annual depreciation of the navy-yard property is about as follows: Temporary structures on shore, 1½ to 5 per cent.; steel floating docks, 3 to 4 per cent.; temporary wharves in warm waters, 3 to 8 per cent.; structural appliances exposed to action of salt water, 5 to 10 per cent.

"It is not surprising, under such conditions, that roofs are leaking, floors uneven, streets unpaved, and buildings depreciating. Unless an adequate appropriation is obtained for necessary repairs to governmental property at this and other navy yards, it may happen, in times of emergency and just when some of these utilities will be most needed, that they will be found in such a state of depreciation that the time required to repair certain structures might be of such an extended nature as to materially interfere with the expeditious preparation of the fleet for sea.

"It is of paramount importance to the efficiency of the fleet that all transportation and handling appliances, and particularly all piers and docks, should be kept in an excellent state of repair and preservation, so that the extreme limit of their possibilities could be utilized when an emergency arises.

"The Board would be derelict in duty did it not frankly point out the absolute necessity of the Depart-

ment's requesting appropriations for allotment 'Repairs and preservation,' which are commensurate with the value of the property to be maintained in an efficient state of preservation."

EXCESSIVE EXPENDITURE FOR THE UPKEEP OF WAR VESSELS.

The severe restrictions that apply in the construction of war vessels in floor space, height, and weight of appliances have no counterpart in designing merchant vessels, and therefore the factor of safety of the material entering into the manufacture of the dynamic appliances of our warships is small compared with the factor allowed on shore.

The breaking of struts and propeller shafts, the cracking of casings and cylinders, the straining of joints, the rupturing of boiler tubes, and the collapse of auxiliaries may therefore be expected to take place on the warship much more frequently than in the merchant service.

Unless our naval stations are developed to an extent commensurate with the increase of the fleet, we will not only have the vessels laid up for months, perhaps years, but the cost of the work may develop into something staggering. Delay in conducting ship repairs involves excessive indirect expense, independent of the military disadvantage connected with carrying on work in this manner.

OUTLYING NAVAL BASES WILL PROBABLY NEVER BE BUILT UP UNTIL HOME BASES ARE DEVELOPED.

Important as it may eventually be to our first line of defense to have outlying naval bases in the Caribbean and in the Pacific, the probability of developing such stations is too remote to be seriously considered until we have at least a sufficient number of fairly equipped home bases on the Pacific as well as on the Atlantic and Gulf coasts. The unsatisfactory results that have been achieved during the past 10 years in the attempt to upbuild Guantanamo and Olongapo shows the disinclination upon the part of Con-

gress to delay developing outlying stations until the more important home demands have been adequately met.

As regards an outlying naval base at Hawaii, do not claims of that locality stand upon a higher and better plane than that of Guantanamo, Olongapo, and Guam. Pearl Harbor should be considered as a home base, and its development be given precedence over that of any other outlying naval station.

No maritime power that ever existed has created great outlying naval bases before establishing efficient home stations. The outlying base is the corollary of the home station, and the failure to recognize this fact has heretofore resulted in the neglect to develop either complete or adequate outlying or home bases. Great Britain, even with its extensive commerce with North and South America, has only semblances of great naval outlying bases at Halifax and at the Bermudas, for she well knew that the development of any great outlying bases near the United States would have resulted in the building of a half dozen great home bases on our own coast. Is it not a fact that the home base stands for national preparedness and safety, while the establishment of the outlying naval base is primarily associated with aggrandizement and territorial expansion? While we may eventually need certain outlying naval bases, their establishment should not precede the complete development of the home stations.

It should be remembered that an outlying station of one nation may be very near the home coast of some possible commercial or rival naval power which may not look complacently on. If we attempt to develop distant bases in the Eastern Hemisphere, will it not justify European or Asiatic nations in establishing such stations in the Western Hemisphere and thus in equity invalidate the Monroe doctrine? We have twice almost gone to war in maintaining that the United States will not permit the acquirement by a foreign government of certain islands in the West Indies as well as of certain possessions on the Pacific.

The cession of Heligoland by England to Germany is also illustrative of the fact that there is a limit to the extent to which any nation will be permitted to retain possessions or develop naval bases, almost within sight of the home shores of a formidable industrial and maritime rival.

National thought and purpose with regard to a suitable time for the development of outlying naval bases has probably been best expressed by Mr. Edward H. Finley whose words upon general naval preparedness seems specially appropriate to this particular feature of the problem.

> "The defensive period should be begun at once. First, a defensive screen must be built as already specified against foreign aggression; and then behind that line of defense the offensive-defensive could be safely and vigorously undertaken. If the order indicated is not observed, our fleet could not later be built up without Europe's permission."

THE BUDGET SYSTEM OF APPROPRIATIONS.

There are those who would even have us believe that by the introduction of a Budget system of appropriation there could be effected economies which would materially reduce naval expenditures. Such a conclusion is a visionary belief not founded upon either fact or experience. It is simply delaying national preparedness to place much faith in the suggestion that by the adoption of a Budget system of submitting estimates any substantial financial benefits would arise. The lumping of certain items in various departmental appropriation acts may prove extremely advisable, since more rather than less latitude in national expenditures should be allowed the Cabinet officers, but such details of expenditures bear no inherent relation to a Budget system. The vetoing of individual items in appropriation acts should be made possible, but such action has nothing to do with a Budget.

At least several hundred thousand dollars have been di-

rectly or indirectly expended during recent years by the National Government in attempting to discredit our existing departmental system of making estimates, and to substitute some scheme of Budget expenditures. The only good that has ever been subserved in the expenditures for experimenting with the Budget system so far as the Navy is concerned, has been in showing that a system of appropriation of the general character proposed by the advocates of the Budget is inapplicable to its business administrations, whether the proposition be viewed from either an economic or military standpoint.

One of the primary advantages claimed for a budget is that it will not only show more in detail how appropriations for national purposes are expended, but that it will obviate the presentation of deficiency estimates. It has been unreservedly stated by the supporters of the budget scheme that naval deficiency appropriations are something unknown to the British Parliament. As bearing upon the reliability of such contentions, it may be stated that the First Lord of the Admiralty, on February 12, 1914, submitted to the House of Commons a naval supplementary estimate (a deficiency measure) for a sum not exceeding two and one-half million pounds sterling, to be granted for additional expenditures for naval purposes.

It is pertinent to state that the British Admiralty has much less reason for submitting a deficiency estimate than our Navy Department, since the Admiralty and War Department officials of Great Britain have authority to take any unexpended balance of one appropriation and divert it to any other authorized purpose. In other words, every Cabinet Minister of Great Britain can, in an emergency, divert an appropriation granted for one special purpose to meet some other obligation where there is likely to be a deficit. Would the Congress of the United States permit the executive departments of the Government thus to divert appropriations, if not to nullify certain acts of the legislative bodies?

As stated by the Admiralty, the underspendings of one estimate can be applied to an overspending of another estimate. In closing the debates on these estimates, the Parliamentary and Financial Secretary of the Admiralty spoke in the House of Commons, on March 4, 1914:

> "With great respect I venture to think that if a stranger had been amongst us today he would almost have come to the conclusion that nobody had ever seen a Supplementary Estimate before. My honorable friends have been turning over the pages of this Supplementary Estimate as if it were a curiously rare and cryptic document which had mysteriously come to light and which had never been seen before. Watching them turning it over I confess I began to lose my identity and feel like a man dug out of some ancient catacomb and brought to life again. That, at any rate, is the impression left on me by the discussion. But we have had four Supplementary Estimates in the last 10 years, and all have been in the same time-honored form. If the form is difficult, if it is obscure, at any rate it is not a novel form; it is not our form, it is a form established by custom."

MILITARY VALUE OF THE COMMERCIAL SHIPYARD.

Important as it is to develop our navy yards, it is also important for national defense that encouragement should be given to the development of our commercial shipyards. These plants are of great value to the Nation, and it is due in considerable part to our industrial resources that our influence as a naval power is seriously taken into consideration. Concerning the worth of the navy yards and private shipbuilding plants, the question may well be asked, is it not America's eventual ability to replace, if not to multiple, her existing war fleet, that after all has done much for our

immunity from attack, and which constitutes one of our most important factors of naval strength?

Encouragement should be offered our armor-plate manufacturers, shipbuilders, and all others engaged in producing war material to compete for Government work. Germany practically subsidized such industrial concerns as Krupp's, in the effort to help that firm lead the world in the production of structural material and heavy guns. It was continuous experimental work and research investigation of the manufacture of munitions of war that has very materially contributed to the distinguishing success of the German Army.

Great Britain, of all countries of the world, has now a full measure of appreciation of the far-reaching military service rendered by the commercial firms engaged in the manufacture of munitions of war. And yet she has very plainly given these manufacturers to understand that the nation's necessity is not to be made the manufacturer's opportunity. She has therefore declared that the nation must regulate control of the extent of the profit that is reaped from the manufacture of war supplies. One of her war measures has been the limitation of the amount of net dividends that shall be paid to the stockholders of such corporations. If, however, such control is justifiable in time of war, it seems but equitable that the commercial firms should be given a fair share of Government work in times of commercial depression. The integrity and patriotism of the manufacturer should not be impeached without cause. Is it not, therefore, probable that foreign powers, when planning campaigns against us, will give serious consideration to the reserve facilities that we have or could develop in the shipbuilding line? As our maritime resources in shipbuilding are lamentably weak along the South Atlantic and Gulf coasts, does there not appear an imperative need that the southern navy yards should be developed to the highest possible state of efficiency?

SHIPBUILDING DURING THE PAST DECADE AN UNPROFITABLE VENTURE.

After the completion of the present war, will it not be necessary that the shipbuilding establishments located on the Atlantic and Pacific coasts shall receive some form of national aid or subvention to make them profitable ventures? During the past 12 years the number of large shipbuilding plants have not increased upon either the Atlantic or Pacific coasts. Shipbuilding has only been kept alive during the past decade by three fairly profitable classes of work—the construction and repair of vessels for the coastwise trade; the extensive building of various types of river and harbor craft; and the carrying on by a limited number of firms of special manufacturing activities. The Cramp Co. find the extensive manufacture of Parsons bronze and the building of water turbines two of the most profitable features of their business. On the Great Lakes the shipbuilding industry, due to some special causes, has a better chance to make the business a fairly safe venture.

The only profitable forms of naval construction carried on for many years at our commercial shipyards have been the building of tugs, gunboats, tankers, colliers, and cruisers. The building of Destroyers, Submarines, and Battleships in America has been, in general, a very unprofitable undertaking. Except in the cases of the Herreshoff Co., the Bath Iron Works, and, possibly, one other firm, the construction of Destroyers has proved disastrous to the shipbuilding industry. Of 15 firms that received contracts for torpedo boats and destroyers, at least half of them went into the hands of a receiver, primarily as a result of the financial losses that accrued from attempting to do such work. It is a conservative estimate, and it is based upon some reliable evidence, that the construction of every single battleship at commercial shipyards during recent years has entailed an average loss upon each vessel of about a half million dollars

to the contracting firms. As regards the construction of submarines, has any firm engaged in the work heretofore found the building of such craft profitable?

The inquiry will naturally be made, Why did the shipbuilders seek any class of naval work at less than the cost of construction? The answer is that in the case of the Submarines and Destroyers the work and the difficulties attending their construction when submitting bids for such work were unknown.

As regards the contracts for the battleships, they provided extensive work at a time when the plants were comparatively slack, and therefore the commercial firms, in order to prevent possible bankruptcy of their establishments, considered it good judgment to temporarily carry on work at a loss.

Then, again, both Shipping and the shipbuilding industry of neutral nations invariably develop into very profitable activities when other maritime nations are at war, for then the demand for ships and ocean rates rise rapidly. In the belief likewise that this nation would enact into law certain legislation that would make shipbuilding a fairly good investment, that the Isthmian Canal would be free to our coastwise trade, and that there would also be an eventual call from South America and China for vessels, the shipbuilding industry was operated for years at a loss. Can more indisputable evidence of this fact be cited than to note that for the 12 years preceding this war the exceedingly depressed value of the common stock of these corporations, together with the certified and sworn annual statements of the officers of the shipbuilding companies concerning their financial operation? The shipbuilder is entitled to a fair profit upon his investment, and it ought to excite neither surprise nor resentment that he now declines to continue to build battleships at a financial loss. Shipbuilding is just as much entitled to reasonable returns as the planting of wheat and cotton.

SHOULD ANY OF OUR NAVAL STATIONS BE ABANDONED?

If measured even by our present naval demands, none of our naval stations should be abandoned or closed. The question is rather one of providing additional dry docks, as well as increased berthing and handling facilities at each of our navy yards. Our deficiency, and not our superfluity of naval stations will forcibly present itself if we ever engage in war.

It is important to note that the navy yards at Portsmouth, N. H., Boston, New York, Philadelphia, and Norfolk were originally located in the heart of those ports and at a time when the commerce of those harbors was comparatively small. Very desirable waterfrontage sites were therefore acquired for navy-yard reservations. Particularly was this the case at Boston, New York, and Philadelphia. Even as early as 50 years ago the railroad and steamship experts of Philadelphia realized that a navy yard at the foot of Federal Street of that city would be an insurmountable bar to the efficient development of the Port, and as a consequence the navy yard was transferred to League Island, a point 5 miles down the river. The new naval reservation is many times the size of the one that was abandoned.

As regards the Boston Navy Yard, it may only be within a decade when the transportational terminal demand of that city may compel thoughtful consideration of the important question as to whether the channel frontage on the Mystic and Charles Rivers, now possessed by the naval authorities, may have to be vacated to meet the transportation and distribution demands of the railroad and steamship lines having their terminals in that city. While the Commonwealth of Massachusetts has already authorized the construction of a dry dock and terminal pier facilities at South Boston to help meet the present needs of the railroad and shipping companies, these projected facilities will probably only satisfy in part the demands of the progressive and rapidly growing commerce of the port.

The Collector of the Port of New York has repeatedly called attention to the congested conditions as regards the commerce of that Port. It requires no prophetic powers therefore to predict that possibly within a decade there may also come a commercial demand, perhaps irresistible, for vacating the present Boston and New York Navy Yards.

Our existing pier-wall and dry-dock facilities at all our naval stations is absolutely inadequate even for present needs, and yet the Navy Department may be called upon to unqualifiedly prove the imperative military-industrial necessity for continuing large naval stations in the heart of the maritime districts of great ports like Boston and New York, where the terminal facilities possessed by the navy yards may better serve the purposes of the nation by permitting them to be acquired by the commercial interests of these great harbors, whose water frontage is now severely taxed to meet the demands of shipping.

The terminal transportation and distributing demands of great seaports like Liverpool, London, Bremen, Hamburg, and Antwerp could not possibly permit the continuance or establishment of naval dock yards in the midst of their shipping districts. Will not this nation be compelled, at an early day, to conserve the entire water frontage of certain ports for maritime purposes? Great as may be the loss to the Navy by vacating important yards like those of Boston and New York, the impelling and growing demands of commerce may ultimately cause such action to be taken. It is the lack of terminal distribution facilities which now materially contributes to the increased cost of food supplies.

The necessity of giving early and thoughtful consideration to the question of enlarging the Portsmouth, N. H., Navy Yard and of developing a new station at Narragansett Bay to provide for the possible vacating of the Boston and New York Navy Yards is a matter that particularly merits the attention of the Navy Department and the Congress.

OUR NAVAL STATIONS ARE NOT COMPLETELY EQUIPPED ESTABLISHMENTS.

For the past generation we have been progressively augmenting the fleet, while pursuing a policy of retrenchment and severest economy in the development of our navy yards. The parsimony manifested as to navy-yard extension has been due to the rapid and incessant attacks of the sensational writers of the daily press and popular magazines, who should have been better informed than to proclaim that a "pork-barrel policy" had dominated the establishment, administration, and operation of our naval stations. Has any naval station ever been established or even its territorial limits abridged except upon the recommendation of some high ranking officer of the Navy?

Time will show that cheese-paring economy in this direction has been at the expense of the fleet, for if the future efficiency of our naval stations has thereby become impaired, the fleet will be the ultimate victim of such a suicidal policy.

Have we not had all the warning we need that behind the troops in the field and the blue jackets on the battleships there must be at least fairly completed navy yards with facilities for employing the army of workmen required to maintain the fleet as the first line of defense?

Our naval stations are not finished works. As a matter of fact for a considerable number of years past it has not contributed to anyone's official reputation or popularity to recommend their extension and development, and such may continue to be the case for a few years to come. The day of awaking upon this question is, however, much nearer than anticipated, for the normal increase in the repair work of the fleet will soon tax the resources of our naval stations.

Is there an officer in the Navy, who possesses accurate knowledge of the industrial and military conditions existing at any of these stations, who now regrets that our navy yards have been developed to the limited extent that has

been effected, and who does not hope that further extensions may soon be authorized? In brief, the outlay incurred for navy-yard development has been as efficiently, wisely and economically expended as have been the appropriations for the battleships.

Unless we immediately change our policy of seeming indifference in developing naval stations, we may within a few years be compelled to undertake this work intensively and therefore expensively. We shall indeed be fortunate if the work will not have to be done at a period following the loss of a portion of the fleet and its personnel. It is the unfinished and undeveloped weapon or appliance which fails in time of emergency. Taken as a whole, the most unfinished and incomplete feature of our naval establishment is our navy yards.

WHAT CONSTITUTES A FLEET?

In the age of sail the floating craft within themselves comprised the fleet. Such, however, is not the case in the age of steam. While the sailing brig could cruise for months with her contained resources, the modern battleship cannot be operated beyond a week under maximum forced-draft conditions. It is only a pleasing fiction that the super-dreadnaught is self-contained as regards repairs. The armored ship does not carry enough armor-piercing and common projectile shells to supply her turret guns with ammunition for an hour.

The Dreadnaught needs underwater eyes in the shape of Submarines, surface eyes in the form of Destroyers, and overtopping eyes of the Aeroplane order; she requires reserve powder, shell, and torpedoes of the value of millions of dollars; she needs a chain of wireless stations to complete her means of communication. It would require pages of manuscript to tell of all requirements. These demands are of the most expensive and extensive character. It is even needful that we should widen and deepen our channels

to our leading seaports in order that the battleship can enter harbor for supplies and repairs.

It is probably beyond the knowledge of any individual to realize in detail the fullness of the auxiliary demands of fighting ships. The more thoughtfully one studies this question, the more appalled he becomes at the extent of these demands.

Is it possible that any program of naval construction can be considered an adequate and well-rounded one which does not allot, or even obligate, for extension of the shore stations, for the development of auxiliary industrial activities, and for improving the channels to our commercial shipyards and naval stations, an appropriation of at least 10 per cent. of the amount proposed for the augmentation of the Navy afloat? Great Britain now finds it necessary to expend millions in providing or extending her naval shore establishments. Not even 2 per cent. of the expenditure involved in carrying out the naval program outlined has been allotted for the development of our naval stations. While no thoughtful officer desires to have the extent of naval construction afloat abridged, there are those who are amazed that substantially no provision appears to have been made for the necessary development of navy yards.

CONCLUSIONS.

The naval program as submitted to the Congress is a satisfactory one only to the extent that it may meet our needs as regards the fleet afloat. This program appears to be substantially based upon the assumption that national safety will be obtained through a decisive battle upon the first line of defense; it practically assumes that our naval shore stations are completed industrial establishments—which they are not.

Neither our strongest possible naval foe, nor even any combination of naval powers, would invite or accept war with the United States without the confident conviction that

their fleet would not only reach our shores but drive our battleships into harbor. As it would be a battered, disabled, and depleted American fleet which would seek the protection of our coastwise defenses, ought not we provide for a contingency which may develop into a verity even before many years elapse?

The failure to adequately develop our naval stations in a manner which will enable them to repair expeditiously a disabled fleet is simply inviting a prolonged naval war, if not national disaster. If for a stretch of 2,500 miles of our South Atlantic and Gulf coasts we neither possess a dry dock that can receive a superdreadnaught nor a crane that can handle the engines, boilers, and turret guns installed on such fighting craft, and if a battered fleet has to wend its way past Cape Hatteras to make for a harbor where it can be docked and repaired, who can maintain that a naval program which only provides floating craft and some reserve ammunition is a well-rounded one?

The building of a dry dock requires a longer period than the construction of a Battleship, and, in view of what is transpiring daily, ought not the probability of the impairment of the dry dock to be considered more likely than the chances of disablement of the Dreadnaught? And yet the program submitted makes no provision whatever for the building of a modern dry dock at any port between Cape Hatteras and the Rio Grande, nor upon the Pacific coast.

Our possible naval foes will scarcely devise strategic plans that ignore the existence of the inefficiently equipped chain of naval stations that we possess on the South Atlantic and Gulf coasts. Not one of these stations has even a suitable magazine for the storage of reserve shell and ammunition. We have been five years trying to procure an efficient floating or stationary crane suitable for the needs of a superdreadnaught. Is there anyone who will maintain that the floating crane at the Boston Navy Yard or the two floating cranes at the Canal Zone are fully satisfactory appliances for naval needs? And yet the projected program does not

provide at any naval station for either a single stationary or floating crane capable of lifting a weight of 250 tons, although our possible foes possess such appliances. The existing equipment and development of our southern naval stations are a reproach to the nation. Nearly everything in the shape of our naval shore stations on the Atlantic is now concentrated between Cape Ann and Cape Henry, and we are told that strategic consideration warrants such action, since the South Atlantic and Gulf coasts can be protected from Guantanamo—a projected location wherein it has not yet been established that an assured water supply can be procured and protected.

As regards the practical value of the ordinary confidential strategic plan, is it not probable that there are in the military vaults of many European powers thousands of discredited and discarded plans which even two years ago were regarded as of the most important and valuable nature by the respective general staffs of Europe, but which today are not worth the paper upon which they are written?

Does this nation need a more timely warning of the fallacy of concentrating industrial activities of either a naval or a military nature than the sad predicament of France in losing, within two months of the declaration of war, about 70 per cent. of her metallurgic production as a result of the capture by Germany of the bulk of the steel foundries of the Republic?

M. Albert Thomas, the master of the forges of France, states—"that the greatest lesson of this war is that, without a definite program for setting one's industries on a war basis, no nation can now safely go to war."

Measured from this standpoint, have any, and particularly the southern navy yards, been planned on a war basis, and should we not, if necessary, sacrifice the building of two battleships to develop our Atlantic and Pacific naval stations in a manner which would give them facilities for rapidly placing in times of emergency our disabled battleships again on the battle line?

The first duty of this nation is to "prepare." No fleet, however, that we can assemble will fulfill its mission if our navy yards and shipbuilding establishments are not immediately developed to a point where they can efficiently, completely, and expeditiously "repair" the disabled and battered battleships. Neither our commercial shipyards nor our naval stations are now in condition to meet the extensive demand of war. It will certainly involve an expenditure approaching the cost of three battleships to efficiently develop all our navy yards and to adequately improve the channels leading to them.

Our needs, also, as regards a reserve of stores, coal, fuel oil, mines, smokeless powder, ammunition, and torpedoes; enlargement of our radio stations, extension of aviation facilities, establishment of a naval research laboratory, together with various other activities, should likewise be concomitantly developed with the augmentation of our fleet.

Is it an excessive estimate, therefore, which calls for 12 per cent. of the amount contemplated for the increase of the fleet afloat, to be expended for the numerous auxiliary purposes connected with the repair, maintenance and operation of the floating craft? Under war conditions it will undoubtedly be shown that we should have expended at least 20 per cent.

No friend of the Navy desires to have the building program for fighting vessels reduced. If, however, the navy yards cannot be developed in any other manner, then a well-rounded plan of naval preparedness demands that the program as regards the augmentation of the floating craft be reduced to a sufficient extent to provide adequate repair facilities for the fleet that we shall build.

In brief, our naval program either appears to be oblivious to the far-reaching influence of industrial operations upon modern warfare, or else it presupposes that our existing industries have been developed and organized sufficient to meet war requirements. It may be, that along certain lines our ordnance facilities have been vastly enlarged, but such

is not the case with our shipbuilding and marine engineering resources.

As regards the industrial conditions in Europe, the output of the Krupp works has been increased 350 per cent. in a year. Great Britain now has over 2,000 controlled establishments for making munitions of war in place of 500 she had last June. The cost of the Zeppelin dirigible has increased threefold since the war. From the port of New York at least $2,000,000 worth of war material is being shipped daily. Is not the man of the hour in Great Britain the man who was the first to realize that the war in which the Empire was engaged was not only a struggle between military forces but a contest as regards the industrial resources of the opposing nation? Does it not appear as if Britain's hope of victory now rests as much in her industrial supremacy as in the vast forces which she has in the trenches? Is it not therefore in contravention of the signs of the times that we have outlined a plan of naval preparedness that substantially ignores the extensive and inestimable military-industrial work that will devolve in war upon our naval stations?

After having been permitted to visit Britain's grand fleet, Mr. Archibald Hurd states, in a quasi-official message to the British public, that, independent of battleships, battle cruisers, cruisers, destroyers, and submarines, England now has in use 2,300 patrol vessels, mine sweepers, etc., together with 2,000 tugs, transports, and supply vessels. Does it not require something of a plant to repair and dock this multitude of auxiliaries in addition to the fighting fleet?

In the confidential report submitted by the general board on July 30 last, concerning naval needs, the closing paragraph is:

> "Increased facilities for the navy yards and shore establishments generally, such as dry docks, berthing spaces, building slips, structural shops, cranes for heavy weights, shop machinery, ammunition, and other storage facilities, civil personnel, etc."

Does not this recommendation call for a heavy expenditure for navy-yard development?

In his message to the Congress, President Wilson declared that the goal of all plans of preparedness should be "national adequacy." To be half prepared is but one remove from not being prepared at all. If we greatly augment the fleet without concomitantly developing our our naval stations, who can maintain that "national adequacy" is being provided for in a naval program which, in substance, makes only for the strength and efficiency of the fighting fleet and ignores the needs of the shore stations which are expected to keep that fleet on the battle line. Does not such a plan for preparedness invite disaster?

AN ALTERNATING-CURRENT SYSTEM FOR ENGINE TELEGRAPH, HELM-ANGLE INDICATORS, ETC.

By Leonard A. Doggett, Associate.

There is a wide field on shipboard for interior-signalling systems, such as engine telegraphs, course telegraph, fireroom telegraph, helm-angle indicators, turret tell-tale systems, gun-firing signals, steering telegraph and smoke telegraph.

To date such systems have usually been operated on two general principles. In the first, the transmitter operates a revolving arm which so arranges the electrical connections that incandescent lamps burn at the transmitter station and corresponding lamps at the indicator or receiver station. These lamps illuminate the desired orders. In the second system a step-by-step motor is used, one type of which is made by Chas. Cory & Son. In this motor the stator is in minature the field structure of a multipolar direct-current machine, while the moving element, which is in the armature space, is a bar of soft iron, bent to have suitable pole shoes. By proper excitation of the field poles singly or in pairs, a moderate number of different positions may be given to the moving element which does the indicating. Other arrangements of rotor and stator, but involving the same principle, have been developed. These systems act with entire satisfaction as long as the number of orders to be transmitted is small. When, however, a turret tell-tale system of the motor type is called upon to show 36 positions in one revolution, the number of wires between transmitter and indicator becomes large, twenty or more. Helm-angle indicators may take ten or more wires between transmitter and indicator.

To reduce the number of wires, therefore, with no sacrifice

in performance seems desirable, especially, as will be pointed out, a number of other advantages come with the system to be described.

For brevity this system will be called the Five-Wire Alternating-Current Telegraph System. This system has been used successfully in connection with the Panama Locks * and on board the Argentine battleship *Moreno*. As installed on the *Moreno*, Mr. H: A. Hornor has briefly described the apparatus in the 1914, October, Proceedings of the American Institute of Electrical Engineers.

During the summer of 1915 Lieutenants C. C. Thomas and S. S. Kennedy, of the Post Graduate Department of the United States Naval Academy, made a study of this system, and their results are here reported, together with some extension of their work.

Preliminary experiments were made with crude apparatus to gain a knowledge of the operation of the system. For this the two similar gramme ring stators, shown in Figure 2, by S_1 and S_2, were prepared. The cores of the stators consisted of 12-pound coils of number 20 B & S soft-iron wire. Around one-third of the core were wound 300 turns of No. 16 B & S copper wire. The remaining two-thirds, being similarly wound, produced a 220-volt three-phase stator winding, shown diagrammatically in Figure1. These stators were connected as shown in Figure 3.

In stator number 1 was placed a laminated core wound with a hundred turns of No. 10 B & S wire. In stator number 2 was suspended a coil of wire wound on a split brass tube. Figure 4 shows the electrical connections and Figure 2 shows a photograph of the arrangement.

If to the terminals A and B of Figure 4, single-phase alternating current is supplied, it will be found that rotor 2 will take up the position of rotor 1 and will follow rotor 2 in all its movements. The original apparatus of Figure 2, although being mechanically crude, nevertheless performed the function of transmitter and receiver sufficiently well to indicate prac-

* "General Electric Review," Supplement to Volume 18, Number 7, July, 1915, page 751.

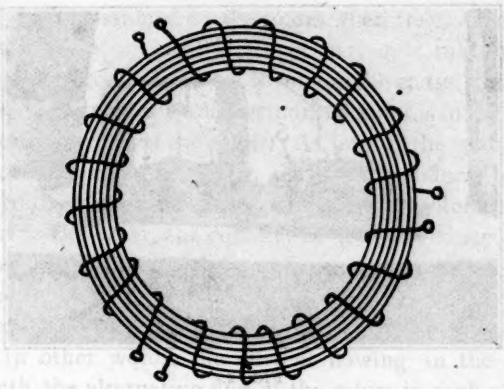


FIGURE 1

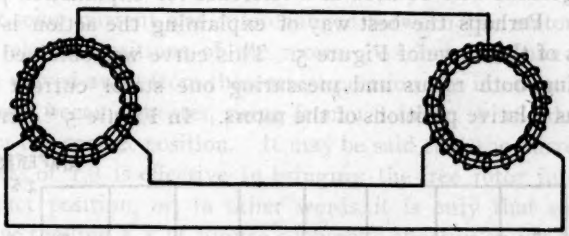


FIGURE 3.

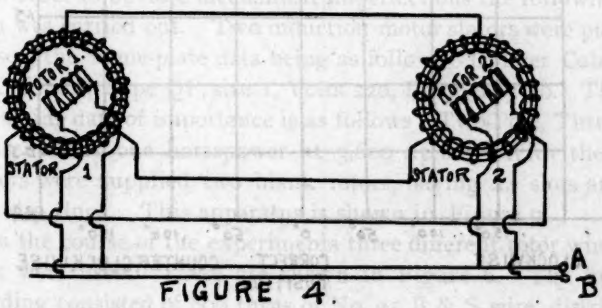


FIGURE 4.

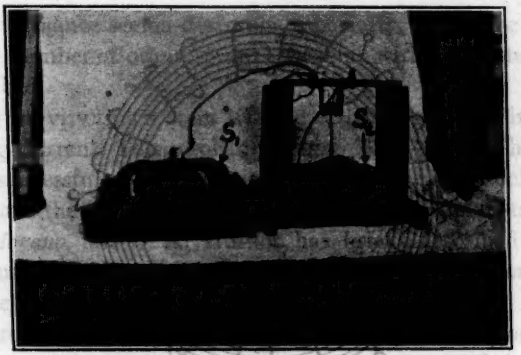


FIG. 2.

tical possibilities with well-made apparatus. Before describing the final apparatus which functioned satisfactorily, the crude arrangement will be used as a reference for explanation purposes. Perhaps the best way of explaining the action is by means of the curve of Figure 5. This curve was obtained by blocking both rotors and measuring one stator current for various relative positions of the rotors. In Figure 5 "correct

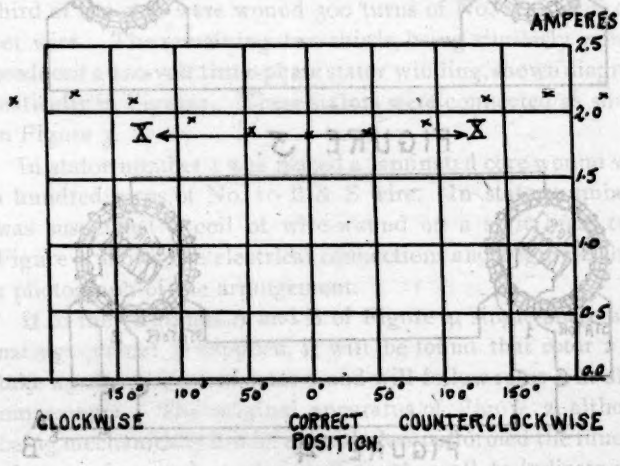


FIGURE 5

position" is that assumed by the rotors when free. One of the fundamental laws of electrical engineering is that induced electromotive forces always act to oppose the cause which sets them up. Here the cause of the induced e.m.f.'s in the stators is the alternating flux of the rotors. As long as the rotors are in the "correct position" and are similar, the induced e.m.f.'s will neutralize each other. As soon as either rotor is moved in respect to the other, the equality of e.m.f.'s induced in the stators no longer exists and the difference at once sends a current through the stator winding which, according to the fundamental law, works to oppose the cause of the residual e.m.f. In other words, the current flowing in the stators reacts with the alternating flux of the rotors in such a way as to pull the free rotor into the correct position. In the apparatus of Figure 2 the rotors were not similar, so that the e.m.f.'s induced in the stators were not equal, with the result that some current was interchanged between the stators for all relative positions of the rotors. As, however, one rotor was moved away from the correct position the interchange of current became greater, rising from 1.8 to 2.2 at 180 degrees from the correct position. It may be said that the current in excess of 1.8 is effective in bringing the free rotor into the correct position, or, in other words, it is only that current above the line XX of Figure 5 which is effective in producing torque.

In order to obviate mechanical imperfections the following plan was carried out. Two induction motor stators were purchased, the name-plate data being as follows: Holtzer Cabot A. C. Motor, Type QP, size 1, Volts 220, Frequency 60. The remaining data of importance is as follows: Two-Pole, Three-Phase, about one horsepower at 3,600 r.p.m. With these stators were supplied two blank rotors, having 24 slots and two slip rings. This apparatus is shown in Figure 9.

In the course of the experiments three different rotor windings were used. These are shown in Figure 6. The first winding consisted of 800 turns of No. 25 B & S wire, distributed in nine coils so as to form a bipolar winding, exactly like

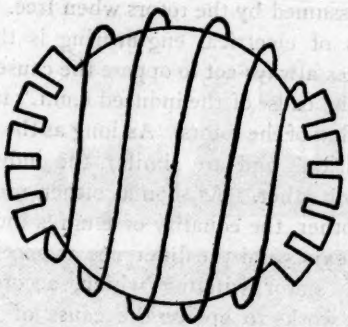


FIGURE 6 c.

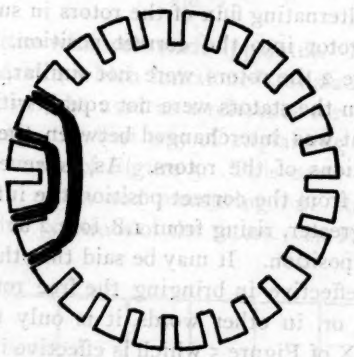


FIGURE 6 b.

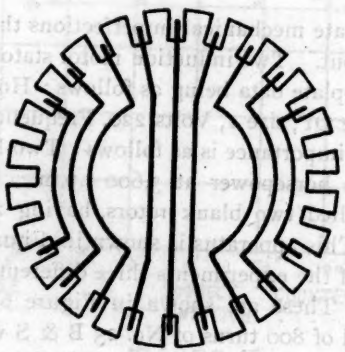
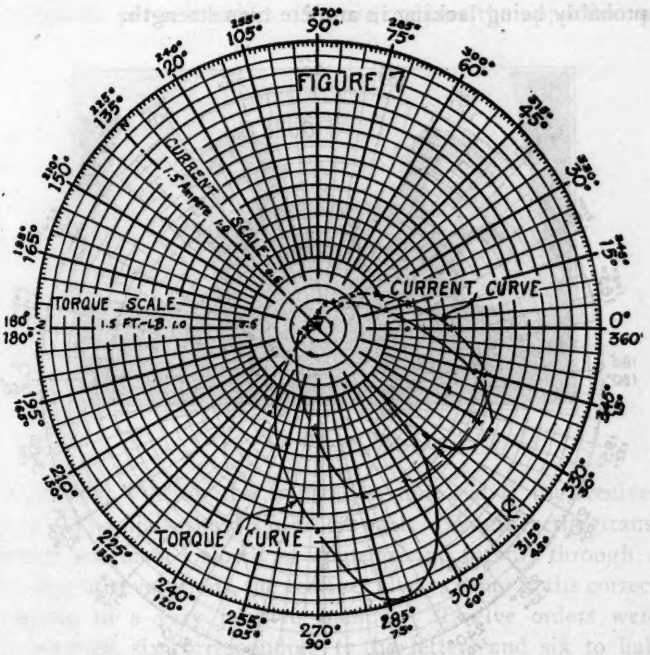


FIGURE 6 a.

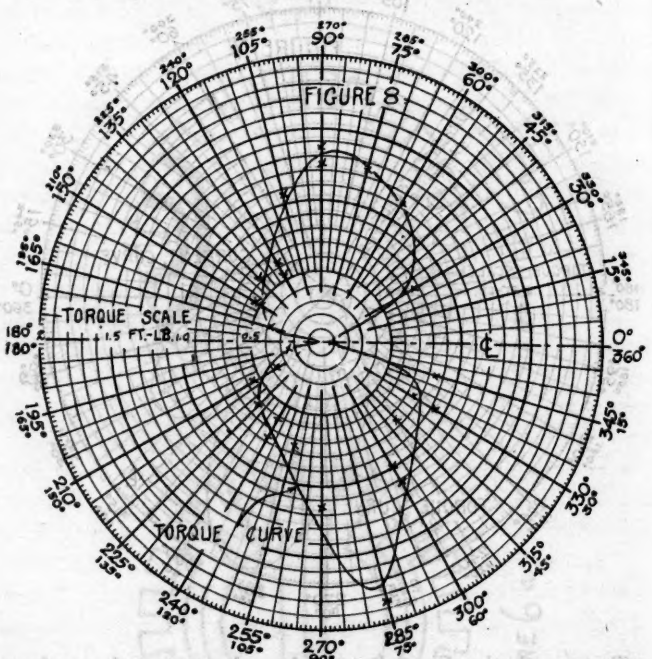
the field winding of large bipolar turbo-alternators. The second winding was of No. 30 B & S wire and was concentrated in four slots as shown in Figure 6*b*. The third winding was of the bobbin type, and is shown in Figure 6*c*. In the winding space provided by chipping off seven teeth on each side of the rotors, were placed 550 turns of No. 24 B & S wire. In the three cases the rotor turns were all in series and were brought out to the two motor slip rings.



Tests with the First Rotor.—In order to test the performance of the apparatus as first set up, two curves were obtained: (1) Average stator current against angular deviation from correct position, and (2) foot-pounds of torque against angular deviation from correct position. These curves are plotted to polar coördinates in Figure 7. From the torque curve of this

figure it is evident that the apparatus is far from sensitive, for the receiver can be moved out of its correct position nearly 90 degrees either to the right or to the left before any considerable restoring torque is encountered. The reason for this lack of sensitiveness is that the rotor winding is so distributed that the pole effect is not sharply defined.

Test with the Second Rotor.—The second rotor of Figure 6 was wound with the idea of concentrating the pole effect in a small, well-defined area. These coils were a complete failure, probably being lacking in ampère turn strength.



Tests with the Third Rotor.—This pair of rotors proved to be much more satisfactory. A curve of foot-pounds of torque against angular deviation from correct position is shown in Figure 8. This curve shows graphically the superiority of the third winding over the first. In Figure 7 the correct

position of the free rotor is at 135 degrees. Not until the 45-degree and 225-degree points are passed does the torque exceed a quarter of a foot-pound, the amount necessary to overcome the friction of the rotor. In Figure 8, however, torque in excess of a quarter of a foot-pound is developed as soon as either the 165 degrees or 195 degrees point is passed, 180 degrees in this case being the correct position. Whereas the first pair of rotors might serve to give three indications per revolution, the third pair of rotors actually gave twelve indications per revolution. Figure 9 shows the apparatus as

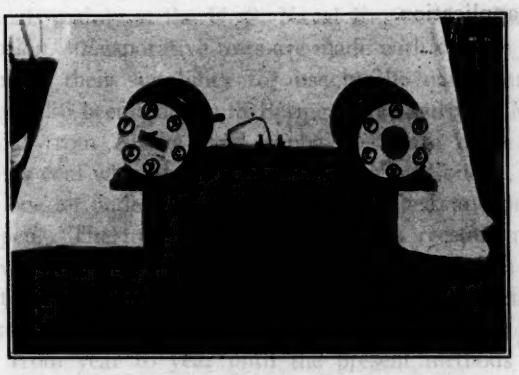


FIG. 9.

finally left, T being the transmitter unit and R the receiver unit. The dials show six orders each. Whether the transmitter was moved slowly or spun around rapidly through a number of revolutions, the receiver always came to the correct position in a very positive manner. Twelve orders were transmitted, six corresponding to the letters and six to half way between the letters.

From the present study of this apparatus two lines of development are suggested. Using the principle here studied, small instruments with light moving parts and jewel bearings to reduce friction could be built to indicate helm angle and position of turrets. For this use transmitters could be rugged

instruments located near rudderpost and turrets. A second application could be for searchlight control, apparatus of about the size and type tested in the above experiments being used. In this case four motors would be needed—one azimuth receiver motor, one altitude receiver motor, one azimuth transmitter motor, and one altitude transmitter motor. The transmitter motors could be placed at the most convenient location, ten wires being necessary between operator's station and the searchlight. It is in this last case that the smooth step feature, that is, the fact that the receiver rotor follows all the motions of the transmitter motor, should find its most useful application.

THE FIRING AND COMBUSTION OF SOFT COAL UNDER NAVAL BOILERS.

By Wm. L. DeBaufre, Mechanical Engineer.

This article on the burning of bituminous coal on Navy standard grates under Navy boilers is based on experience in testing coals mainly at the U. S. Naval Engineering Experiment Station. Evaporative tests are made with various coals to determine their suitability for use in the naval service. Tests have also been made with Bering River and with Matanuska coal from Alaska; and annual tests are made with Pocahontas coal which has been stored at New London, Connecticut, one lot under water, one lot under shed, and one lot in the open. The fifth annual test has been recently made. Successive annual improvements in the economic results were noted after storage for one, two and three years, respectively. These improvements were due to better methods of firing adopted from year to year until the present methods were developed, which are believed to yield about the maximum capacity and economy, that a given coal is capable of producing under a naval boiler.

Coal for naval use consists of from 90 to 95 per cent. of combustible matter and from 10 to 5 per cent. of ash. The ideal method of firing such fuel would consist simply in spreading the coal uniformly over the fuel bed within each of the fire doors in succession. With the burning of the combustible matter, the remaining unburnt material would sink in the fuel bed as fresh coal is spread on top. Such a fuel bed would evidently contain the original per cent. of ash in the topmost layer and larger percentages in the layers beneath. In the bottommost layer just over the grate there would be 100 per cent. of ash, which under ideal conditions would fall freely

through the grate into the ashpit. Under such ideal conditions the only firing tools necessary are the shovel to throw in the coal and the hoe to haul out the ashes.

Unfortunately, other tools are necessary to overcome defects that occur in the practical firing of soft coal. The fuel bed does not burn and sink in uniform layers. It is not of uniform porosity, and the greater quantity of air which naturally flows through a place of low resistance causes more rapid combustion there than in the denser portions of the bed, thus producing a "hole" in the fire. Very few if any firemen are so skilled that they can keep these "holes" filled up and the fuel bed even simply by throwing the coal in the proper places with a shovel. It is therefore necessary to use a light hoe or rake to level the fire.

Also, with marine boilers, the yellow flames so fill the combustion space that it is practically impossible for the fireman to see the condition of the fire on the back half of the grate. It is therefore necessary for him to run a light hoe over the back of the fire to determine its thickness as a guide for coaling, in addition to filling up whatever "holes" may be present. However, the fires should not be disturbed any more than is absolutely necessary to accomplish the desired objects, and the more skillful a fireman is with the shovel the less will he need to disturb the fuel bed with a hoe or rake.

Most coals used in the naval service tend to cake and thus form a solid crust over the fire. Cracks start in this crust through which the air passes, and crevices are formed by the crust burning at the edges of these cracks. Such crusts should not be allowed to form to any extent. While they can be largely prevented by firing smaller quantities of coal at more frequent intervals, they should also be kept broken up by the use of a light hoe.

The hoe is likewise necessary when the coking method of firing is adopted. The coal is thrown onto the front part of the grate only and the volatile matter allowed to distill off, when the resulting coke is pushed back with the hoe over the fuel bed. This method of firing, however, is not considered as

conducive to good results as the alternate or consecutive spreading method; that is, spreading the coal over the entire fire in the furnace doors alternately or consecutively. (The word alternate applies, of course, to two doors only, and the word consecutive is proposed for the same method applied to a furnace with three or four doors.)

Skill in the use of the shovel supplemented with the light hoe or rake will keep the top of the fuel bed in proper condition. Additional tools are necessary for the bottom of the fuel bed, especially with stationary grates as used in the Navy. For those who are not familiar with the Navy standard grate, a photograph of a double grate bar is shown in Fig. 1. On

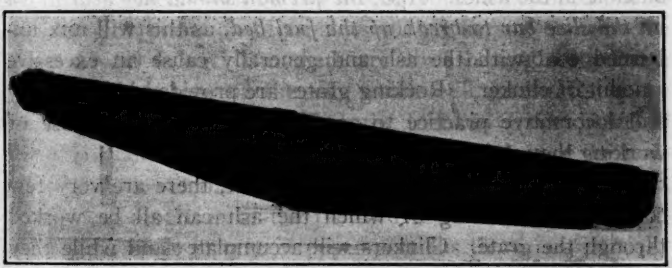


FIG. 1.—NAVY STANDARD DOUBLE GRATE BAR.

this kind of grate ash will accumulate; and, with most coals, part of the ash will fuse together to form clinkers. There are very few coals with an ash-fusing temperature so high that no clinkers are formed.

The ashes that do fall between the grate bars into the ashpit contain combustible matter in addition to the ash. The ash that remains above the grate, whether it fuses into clinkers or not, shuts off the air supply to the combustible above. A fire tool must therefore be employed to work this ash through the grate. If a hard clinker is formed which tends to stick to the grate, a sharp-pointed slice bar is required to pry it loose and break it up. But if the clinker formed is very friable, or if simply ashes accumulate, a better tool for the purpose is a bar

with a flat chisel-shaped end. A pricker bar may also be conveniently used under these conditions.

Neither the slice bar nor the pricker bar should be used indiscriminately, but should be moved back and forth over the grate where the ash or clinker is located just sufficeintly to work it into the ashpit. The location of the accumulation of clinker or ash is indicated by the shadows or dark spots on the floor of the ash pan. Its removal is shown by the brightening of the dark spots. The brightening, however, will generally not occur as soon as the ash or clinker is removed, because the combustible above has been deadened by lack of air. Too much working of the slice bar will increase the loss of combustible in the ashes. Also, *the fireman should not bear down on the slice bar to break up the fuel bed,* as this will mix unburned coal with the ash and generally cause an excessive amount of clinker. Rocking grates are provided in stationary and locomotive practice to accomplish the same purpose of working the ash and clinker through the grate.

But even with the use of the slice bar, there are very few coals in the burning of which the ash can all be worked through the grate. Clinkers will accumulate; and while they may be kept broken up in many cases by the use of the slice bar, they must eventually be removed through the furnace door. The cleaning intervals adopted will depend upon the quantity and character of the clinkers, the variation of the load upon the boiler, the sequence of watches of the firemen, and the necessary routine of removing ashes from the ship.

For a thorough cleaning, the fire within the furnace door selected should be allowed to burn down slightly. At the time of cleaning, the combustible in the top of the fuel bed is winged over onto the fire at either side, using a hoe and a slice or special flat bar. The bottom of the fuel bed containing the clinkers is then hauled out of the furnace with a heavy hoe. The fire is now winged back onto the bare grates, leveled, and built up to standard thickness. Before cleaning, it is preferable to build up slightly the thickness of the fuel bed in the doors that are not cleaned. Also, under steady load conditions, it is

better to clean the fires successively at regular periods unless the routine of the removal of clinkers and ashes prevents. Troublesome clinkers may be removed between the regular cleaning periods by using the tool called the devil's claw.

In working the top and bottom of the fuel bed, it has been emphasized that the fuel bed as a whole should not be disturbed any more than necessary. This is done in order to prevent as far as possible the formation of clinkers. Clinkers are due to the fusing together of particles of ash. The temperature at which the ash fuses depends mainly upon the constitution of the ash in the coal. When the fusing temperature is high there is no trouble with clinkers, no matter how the fires are worked. With a low fusing point it is impossible to prevent the formation of troublesome clinkers, even with the most careful firing. But there are coals having a moderate fusing temperature of the ash which will give satisfactory results by careful firing. When the fuel bed is not disturbed the ash accumulates only in the lower portion where the temperature is lower than in the upper portion of the fire. Also, in the lower portion of the fuel bed there is an oxidizing atmosphere, while in certain upper portions there is a partially reducing atmosphere, and the fusing temperature of ash is less in a partially reducing than in an oxidizing atmosphere.

Thick fires especially produce an atmosphere conducive to the formation of clinkers. The fires should therefore be carried as thin as is consistent with the draft and with good combustion, as measured by the constitution of the flue gases. The better the condition in which the fires are maintained, the thinner can they be carried with good combustion with the same draft.

Draft is the difference in pressure between the air in the boiler room and the flue gases at the point considered. Furnace draft, for example, is the difference between boiler-room pressure and the pressure of the furnace gases just above the fire. Stack draft is the difference in pressure between the boiler room and the flue gases at the base of the stack. The difference between the furnace draft and the stack draft is

therefore the difference in pressure between the gases in the furnace and at the base of the stack. It is evident that this difference is equal to the frictional resistance overcome in forcing the gases through the passes and between the tubes of a water-tube boiler, for example. The furnace draft is equal to the frictional resistance encountered in forcing the air through the fuel bed. The former resistance will vary with the volume of the flue gases, but the latter will also be affected by the condition of the fuel bed.

To force the same quantity of air through the fuel bed a greater resistance is encountered if the fuel bed is thick, if a crust of unburnt coal is formed over the top, if ashes or clinkers clog up the grate at the bottom, or if the sizes of the coal particles in the middle of the fuel bed are such as to pack closely together into a dense mass. The last mentioned cause of increased resistance depends only upon the character of the coal, but the first three are largely within the control of the fireman. Looked at in another way, if the fireman allows the coal to crust over the top, or if he allows clinkers and ashes to accumulate at the bottom, or if he allows the fire to become too thick, the air supply will be cut down unless "holes" are allowed to form in the fuel bed. These assumptions are based upon constant draft conditions; if the draft is raised the air quantity is increased, with the other conditions remaining constant.

Now, it is the air which furnishes the oxygen necessary for combustion. In passing through the ashpit the air is somewhat heated by radiation. It is further heated in passing over the grate bars, which it thus keeps from being overheated. In the fuel bed it is still further heated until a sufficient temperature is reached to cause the oxygen to combine with the hydrogen in the fuel to form superheated steam (H_2O); with the sulphur to form sulphur dioxide (SO_2); and with the carbon to form carbon monoxide (CO), which then burns to carbon dioxide (CO_2). The nitrogen (N_2) of the air passes through unaffected, and oxygen (O_2) is found in the flue

gases because an excess quantity of air over that theoretically required for combustion is always admitted.

Hence the flue gases contain superheated steam or moisture (H_2O), sulphur dioxide (SO_2), carbon dioxide (CO_2), carbon monoxide (CO), oxygen (O_2), and nitrogen (N_2). There may also be present particles of partially-burned coal which will be deposited on the tubes and baffles as soot or carried out the stack as smoke. The flue gases may contain unburned hydrocarbons distilled from the coal if the air supply and temperature are not sufficient to cause their combustion.

The presence of superheated steam (H_2O) in the flue gases represents a loss. The loss is equal to the heat required to raise the temperature of the moisture in the air to the temperature in the stack, and the heat necessary to evaporate and superheat to stack temperature the moisture in the coal, that formed from the hydrogen present, the water added to the coal, and that put into the ashpit. The water added to the coal and the water put into the ashpit are the only items under the control of the fireman. Water may be added to coal to allay the dust in the fireroom or to reduce the loss of fine coal up the stack. No data are at hand to show the reduction in soot and smoke by adding water to the coal. Water should not be used in the ashpit unless it is necessary to keep the grate bars cool, to prevent troublesome warping of the ash pan, or to reduce clinker formation. Clinkers are sometimes reduced by blowing steam into the ashpit.

The amount of sulphur dioxide (SO_2) present simply depends upon the quantity of sulphur in the coal, and is very small in coals for naval use. Sometimes a portion of the sulphur present remains in the ashes or clinkers as sulphur trioxide (SO_3).

The presence of unburned hydrocarbons depends partly upon the volume of the combustion space available, the manner of firing, and the proportion of volatile matter in the coal. If the combustion space is small, the hot gases containing unconsumed hydrocarbons are liable to reach the comparatively

cold surfaces of the boiler and be cooled below the ignition temperature before they are completely burned. Also, if large quantities of coal are fired at a time, the large volumes of hydrocarbons distilled off are liable to lack sufficient air for combustion. If the coal contains a large percentage of volatile matter, the same condition is liable to obtain with careful firing. This fact is one reason why high-volatile coals are not suited for combustion in marine boilers where the combustion space is necessarily limited. From the fireman's standpoint coal should be fired in small quantities to prevent a loss due to unburned hydrocarbons. If necessary to increase the thickness of the fire, this should be done gradually, and not by throwing a large quantity of fresh coal on the grate at one time.

With medium or low-volatile coal and fair firing, hydrocarbons are rarely present in the flue gases, even with the limited combustion space in marine boilers. It is, therefore, customary to omit their determination from the analyses of the flue gases. Neither the percentage of moisture (superheated steam) nor of sulphur dioxide is determined, but the analysis is limited to the measurement of the proportions by volume of carbon dioxide (CO_2), carbon monoxide (CO), and oxygen (O_2) present, the nitrogen (N_2) being found by difference. For routine conditions of service it is necessary to determine the percentage of CO_2 only as a guide to the condition of the fire. Recording CO_2 machines are built for this purpose.

The four quantities, CO_2, CO, O_2, and N_2, are closely related to the air supply. By perfect combustion with just the right quantity of air there would be in the flue gases about 20 per cent. of CO_2 and 80 per cent. of N_2 by volume. With air in excess of the theoretical quantity, O_2 is also present; the N_2 remaining approximately 80 per cent. of gas and the CO_2 and O_2 together being equal to about 20 per cent. The percentage of O_2 is an indication of the amount of excess air. Unless sufficient excess air is admitted the flue gases will contain CO. The presence of CO represents a loss due to incomplete combustion; but the presence of O_2 with the correspondingly lower proportion of CO_2 also represents a loss due

to excess air. When one loss is increased the other is decreased, and the combustion conditions should be adjusted until the sum of these two losses is a minimum. From experience at the Naval Engineering Experiment Station, it is believed that for naval boilers the minimum loss will correspond to a CO_2 percentage between 12 and 12.5 per cent.

The analysis of the flue gases is generally made at the base of the stack. Therefore, part of the excess air present is due to infiltration through the boiler casing. Before the efficiency of the combustion can be properly judged, care must be taken to insure that the casing is airtight. Excess air is also admitted when the furnace doors are opened for coaling or leveling; hence these operations should be done quickly. However, undue haste should not prevent proper placing of the coal and leveling, for "holes" left in the fire during the intervals between the above operations will admit more excess air than the doors open for a few additional seconds.

The most economical results are obtained when the whole fuel bed is kept in a uniform condition. Then the desired percentages of carbon monoxide and oxygen, and the corresponding percentage of carbon dioxide (CO_2) which has been found best by trial, are kept constant by regulating the thickness of the fuel bed. If the CO_2 becomes low it means that too much excess air is passing through the fuel bed, which is therefore too thin. The thickness should be built up gradually. If the CO_2 percentage of 12 per cent. because the layer of clinker cessive due to lack of air. The fuel bed is too thick and should be allowed to burn down somewhat by firing slightly less coal until the desired CO_2 percentage is obtained.

The proper thickness will be between 4 and 8 inches, depending upon the draft required to burn the requisite number of pounds of coal per square foot of grate surface per hour and also upon the clinkering properties of the coal. A clinkering coal should be burned with a thinner fire than a good coal. It may be impossible with a clinkering coal to attain a CO_2 percentage of 12 per cent. because the layer of clinker over the grate will shut off the flow of air through the fuel

bed, and the consequent greater draft in the furnace will result in increased air leakage through fire doors and setting.

If the fuel bed is not in a uniform condition there will be an excess of air at some places and a lack of air at others, resulting in a total loss larger than would obtain with uniform conditions. To get maximum efficiency in combustion it is therefore necessary to place the coal properly with the shovel, to level and break up the crust with a hoe, and to keep the grates free of ashes and clinkers with a slice bar. By this method, also, the maximum capacity is obtained from the boiler for the given draft because the resistance through the fuel bed is kept low.

The stack draft should be regulated according to the load on the boiler. Regulation by the damper is better than regulation by opening or closing the ashpit doors because less excess air is liable to be drawn through the boiler casing by the former than by the latter method. When the load suddenly increases, the combustion is often intensified by the firemen cracking the fuel bed by lifting it slightly with the slice bar. With a coal that tends to clinker, this should be resorted to only in the last extremity and should be carefully done to prevent breaking up of the fire.

The furnace draft may be used as a guide in working the fires. If its value be noted for a given stack draft when the fires are in their normal condition, a rise in the furnace draft will denote an increase of the resistance through the fuel bed, and a drop will denote a decrease. The increase may be due to the fuel bed becoming thicker or clogged with ashes and clinkers at the bottom or covered with a coal crust at the top in case the coal cakes. The decrease will be due either to a thin fuel bed or to holes in the fire. The furnace draft also drops whenever the furnace doors are opened.

There is no practical method of firing, however, with which some non-uniformity in the fuel bed does not occur. When the doors are coaled in succession, part of the fire will be covered with fresh coal while the remainder is bright. This condition is considered more conducive to perfect combustion than if all

doors were coaled at the same time, for in the former case the heat from the bright portion of the fire helps to raise to the ignition temperature the volatile matter distilled from the coal just thrown in the furnace. This method is also considered better than throwing the coal on the front part of the grate, allowing it to coke, and then pushing it back over the whole grate.

The coaling should be done as nearly uniformly as possible, throwing on the same number of shovelfuls each time. The firing intervals should be selected so that each charge will not exceed 30 to 50 pounds of coal (3 to 5 shovelfuls). Then the fireman can slightly increase or decrease the quantity as he finds his fires getting thinner or thicker. He will sometimes find the fire within one door consuming more coal than in the other doors. This condition can generally be traced to a thinner fire which allows a larger quantity of air to pass through to intensify the combustion. If necessary, however, the fire in that door should be kept a little thicker than in the other doors in order to equalize the rates of combustion.

If the coal is fired in small quantities at uniform intervals, and the fires are otherwise maintained in good condition, the smoke formed will have the minimum density for the boiler in use and the coal fired. The lack of head room for naval boilers limits the height of the heating surface above the grate and the combustion space. The coal particles carried along by the gases are therefore cooled by the heating surface below the ignition temperature before they have time to burn completely. Smoke consequently forms. With high-volatile coals a dense black smoke is continually emitted from the stack, but with medium or low-volatile coals smoke should occur for a few seconds only after coaling.

Based upon the above considerations, there has been adopted the following method of firing coal during evaporative tests at the Naval Engineering Experiment Station: The draft at the base of the stack is maintained constant at 0.750 inch water, for this stack draft has been found to burn about 25 pounds of coal per square foot of grate surface per hour, with a fur-

nace draft of about 0.3 inch water. The boiler is allowed to generate as much steam as it can under a steam pressure of 300 pounds gage, which is kept constant by regulating a throttle valve on a bleeder line through which a portion of the steam generated is discharged to a surface condenser. A firing interval of five minutes has been selected, resulting in a charge of about 40 pounds of coal being fired at a time. The furnace has three doors, so that the interval between firing the successive doors is one minute and 40 seconds. The fires are leveled at five-minute intervals, and in the order in which the doors are coaled: When door one is coaled, door two is leveled; when door two is coaled, door three is leveled; and when door three is coaled, door one is leveled. The slice bar is used at hourly intervals, but only if dark spots appear in the ash pan. The cleaning interval is twelve hours. With three furnace doors and firemen's watches of four hours on and eight hours off, the above interval results in the cleaning of one fire in each watch.

Samples of the flue gases from the base of the stack are collected during half-hourly intervals and immediately analyzed for CO_2, O_2, and CO. Analyses for hydrocarbons are made several times during each test, but no appreciable percentages have been found with coals tested to date. A CO_2 machine is also in continuous operation. The firemen are directed to so regulate the thickness of the fires as to maintain the constant percentage of CO_2 which yields the best results. This percentage has been found to be in general about 12 to 12.5 per cent., corresponding to a fuel-bed thickness of approximately 6 inches. Leakage of air through the casing is prevented by carefully plastering before the test all possible cracks with asbestos cement.

THE FRACTIONAL DISTILLATION OF LUBRICATING OILS.

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The physical and chemical properties of all lubricating oils are thoroughly investigated at the Naval Engineering Experiment Station before they are recommended for use in the service. Research work, dealing with physical and chemical methods which give more accurate information as to the properties of lubricating oils, is constantly under way. One of these investigations which is believed to be of general interest is herein described.

OBJECT OF INVESTIGATION.

The object of this investigation was to devise a fractional distillation method for separating a mineral lubricating oil into its component light oils, intermediate oils and heavy oils, and to investigate the properties of these light, medium and heavy oils. Such a method had to meet the following requirements:

(*a*) It must not decompose any of the component oils.

(*b*) By reuniting the component oils the original compounded oil should be obtained without appreciable change in its physical properties.

(*c*) Double fractionation should not decompose any of the oils or appreciably change their physical properties.

(*d*) The method must give concordant results with the same oils.

(*e*) It must be a laboratory method capable of operation in a reasonable length of time and without danger to the operator.

(*f*) The fractions of an oil must be of sufficient number and of volume great enough to allow the determination of their physical properties.

The investigation of the properties of these fractions of oils should yield:

(*g*) Accurate information of the quality of mineral oils used to compound the original oil.

(*h*) Should show the presence of undesirable light or heavy oils and the quantity present.

(*i*) Should enable us to forecast the relative stability of oils when used in the service.

(*j*) Should give us data which would clearly point out to us the most logical path to be taken to improve the quality of lubricating oils, thereby increasing the life of the oil when in use and decreasing engine troubles arising from the use of inferior oil.

(*k*) Should show the different properties of asphalt and paraffine-base oil fractions in the relation between viscosity, flash point, specific gravity and volatility.

(*l*) Should point the way to future research work in lubricating oils.

SELECTION OF METHOD.

The literature on the testing of lubricating oils gave no assistance or information on the fractional distillation of these oils. In selecting a method four possible modes of procedure were considered. They were:

(*m*) Fractionally distilling and condensing the oil under vacuum.

(*n*) Fractionally distilling by the aid of heat and heated inert gases.

(*o*) Fractionally distilling by the aid of heat and violent agitation of the oil.

(*p*) Fractionally distilling by the aid of heat and superheated steam.

FRACTIONAL DISTILLATION BY THE USE OF HEAT AND A VACUUM.

When an oil or liquid is placed in a flask, a vacuum created and heat applied, it is found that the oil or liquid will vaporize at a much lower temperature than it would if the vacuum did not exist. It has been found that the oils constituting lubricating oils will distil without decomposition under the above conditions. A United States patent was granted the Vacuum Oil Co., of Rochester, N. Y., on this process. This method was not considered a feasible laboratory method for the reason that unless the condensation of the oil vapors was perfect some of the lighter oils would be lost; also, because it was desired to work with glassware in order to closely watch the distillation. Distilling in glassware under a high vacuum was considered too dangerous for a routine laboratory method.

FRACTIONALLY DISTILLING BY THE AID OF HEAT AND HEATED INERT GASES.

The procedure of this method consists of heating inert gases such as carbon dioxide or nitrogen to about 380 degrees F. and conducting them into the oil heated to about the same temperature. It has been found that the oil will then distil 250 F. below its ordinary distilling temperature and without decomposition. This process is covered by U. S. Patent 848,903, Art of Distilling, granted to Heinrich Hirzel, of Leipzig-Plagwitz, Germany. He explains his process in full on page 2, lines 85 to 107, of U. S. Patent 848,903. This process was tried out in the laboratory of this station using heated carbon dioxide, and as far as the distillation was concerned it was found to do all the inventor claimed for it; but it was found impossible to entirely condense the oil vapors mixed with the inert gas, and the method had to be abandoned. The resistance of oil vapors to condensation when mixed with inert gases, and also the greatly reduced temperature at which oil vaporizes when exposed to inert gases are worthy of note.

DISTILLING BY THE AID OF HEAT AND VIOLENT AGITATION OF THE OIL.

No reference in any literature could be found which treated of this procedure for distilling lubricating oil. Several years ago I performed some simple experiments in which I mixed a light oil with a heavy petroleum oil, and on heating the mixed oils in a flask which was violently agitated, it was found that the light oil distilled off at much lower temperature than it would have if not agitated. The apparatus required for using the above method in this test would be too complicated, but it is here noted to show that in all probability when oil is used as a lubricant and violently agitated conditions arise which greatly increase the volatility of the oil.

FRACTIONALLY DISTILLING BY THE AID OF HEAT AND SUPERHEATED STEAM.

After considerable experimentation, in which the construction and size of each part of the apparatus were changed several times, a successful method using heat and superheated steam was evolved. This method is considered to be especially applicable for the purpose of this test, since the process of distilling lubricating oils with superheated steam is practiced by a large number of lubricating-oil manufacturers.

APPARATUS USED IN INVESTIGATION.

Plate F is a photograph of the fractional distillation apparatus partly dismantled. Plate G is a photograph of the fractional distillation apparatus connected up for test. The apparatus consisted of a steam boiler, steam superheater, oil still and water condenser, connected up and fitted with thermometers as shown in photographs.

METHOD OF PROCEDURE.

Five hundred and forty grams of the oil to be tested was weighed in the round-bottom flask of the oil still and the appa-

PLATE F.—TEST 555.

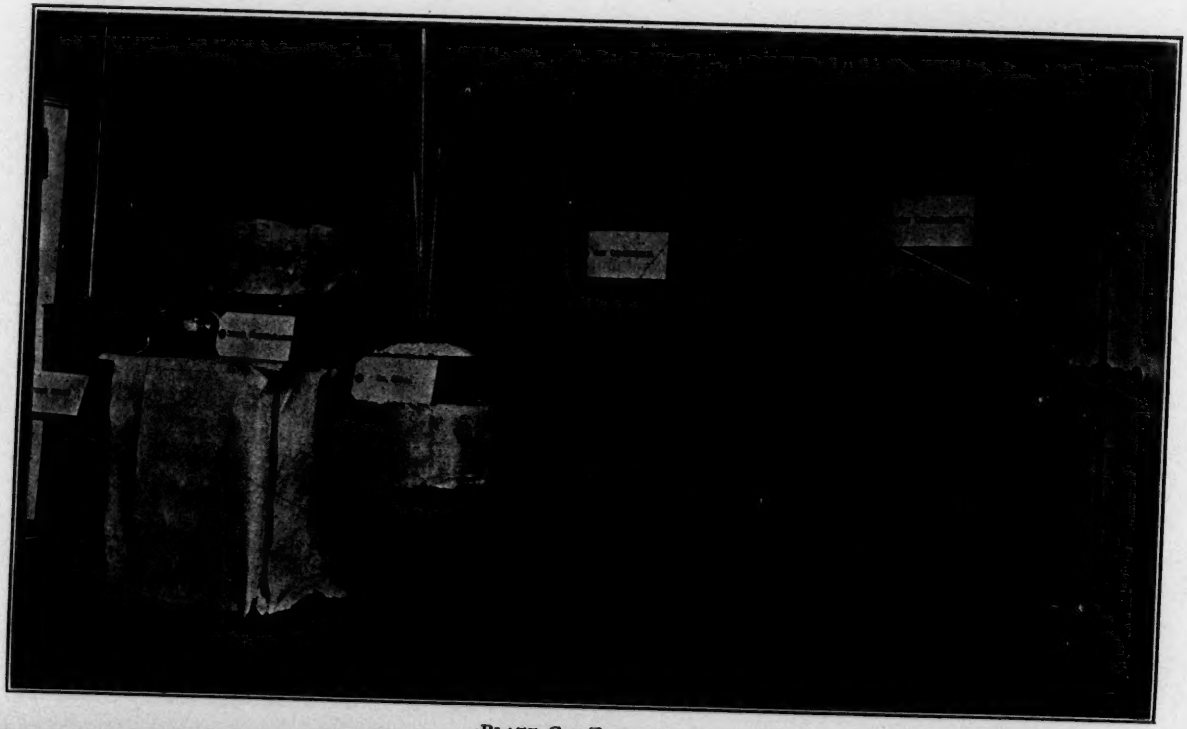


PLATE G.—TEST 555.

ratus connected up as shown in Plate G. The gas burners under the steam boiler and oil still were turned on and lighted. Electric heater was turned on at low heat. When steam was generated in the boiler the heater was turned on full, and the superheated steam allowed to escape through the by-pass between the superheater and the oil still until the steam showed colorless and until the oil in the still reached a temperature of about 380 F. When a colorless, steady stream of steam issued from the by-pass, and the oil was at a temperature of about 380 F., the by-pass was closed and the steam forced through the hot oil in the still. The temperature of the superheated steam and the oil was now slowly raised, the rate of heating being governed by the rapidity with which the oil volatilized. Both of these temperatures were found to be very easily regulated. The volatilized oil and steam from the still first entered the air condenser, and the vapors which passed through the air condenser were all condensed in the water condenser. Approximately three-quarters of the distilled oil was recovered in the receiver of the air condenser; the remaining quarter was recovered in the separatory funnel of the water condenser. The water and oil from the water condenser separated readily in the separatory funnel; the water settled to the bottom and was drawn off and measured. The oil condensed in the air condenser was practically de-hydrated. This was due to the fact that the oil was condensed to a liquid state at a temperature well above the boiling point of water, and also to the fact that the condensed oil in the air condenser flowed by gravity against a very hot stream of superheated steam and oil vapors in leaving the condenser. The oil flowing from the air condenser through the glass U was cooled by allowing the lower part of the U to dip in a beaker of cold water.

The oil was divided by this method into usually four or five fractions, one fraction always remaining as a residue oil in the glass flask of the oil still. The fractions were made as nearly equal in weight as possible under the conditions; they

PLATE G.—TEST 555.

were then dried in a dessicator over calcium chloride or sulphuric acid until perfectly clear and free from moisture. The heating of the oil still was very uniform throughout, as shown by the slight difference in temperature between the oil in the still and the vapors leaving the still. In fractionally distilling a lubricating oil the following data were taken:

(*q*) Temperature of steam, oil and vapor, at the beginning of the distillation of each fraction.

(*r*) Temperature of steam, oil and vapor, at the end of the distillation of each fraction.

(*s*) Time required to distil each fraction.

(*t*) Quantity of water condensed with each fraction.

(*u*) Weight of each fraction.

All temperatures were expressed in degrees Fahrenheit, and no correction was made for exposed stem.

RESULTS.

Origin of Material.

All lubricating oils used in this test were obtained from the Oils Laboratory, and had been previously received for Station investigation. They were representative samples of oils used in the service and all of them are on the acceptable list for use in the Navy. They were all straight mineral oils with the following classification and origin:

Laboratory No. of oil.	Classified as—	Base.
No. 700,	Light forced-feel oil,	Asphalt.
701,	Medium forced-feed oil,	Paraffine.
702,	Medium forced-feed oil,	Asphalt.
703,	Medium forced-feed oil,	Asphalt.
704,	Medium forced-feed oil,	Paraffine.
705,	Light forced-feed oil,	Paraffine.
706,	Ice-machine oil,	Asphalt.
707,	Ice-machine oil,	Paraffine.

Name of oil.	Fraction.	Per cent. of total by weight.	Time taken to distil.	Temperature of distillation, degrees F.						C.c. of water condensed.
				Beginning.			End.			
				Steam.	Oil.	Vapor.	Steam.	Oil.	Vapor.	
No.			*Min.*							
(1) 700	A	17.38	45	390	390	380	500	480	470	280
	B	22.32	24	500	480	470	510	500	490	175
	C	20.10	25	510	500	490	530	524	510	185
	D	22.78	30	530	524	510	550	540	522	275
	E	17.42	Residue oil.							
(2) 700	A	19.14	35	390	390	380	500	480	470	250
	B	19.70	20	500	480	470	510	500	490	155
	C	18.34	20	510	500	490	540	530	520	160
	D	19.00	25	540	530	520	550	540	530	200
	E	23.82	Residue oil.							
(3) 701	A	18.31	60	420	410	400	530	530	415	400
	B	18.38	35	530	530	515	560	554	530	225
	C	18.49	40	560	554	530	585	580	550	250
	D	19.64	33	585	580	550	605	600	564	300
	E	25.18	Residue oil.							
(4) 701	A	23.68	75	440	440	420	535	530	512	415
	B	43.06	85	535	530	512	600	600	566	475
	C	33.26	Residue oil.							
(5) 702	A	27.23	55	390	380	370	520	515	510	460
	B	24.17	25	520	510	515	550	535	525	180
	C	28.39	45	550	535	525	560	550	535	350
	D	20.21	Residue oil.							
(6) 703	A	24.39	50	400	390	380	500	500	500	400
	B	22.30	35	500	500	500	530	525	525	225
	C	24.81	30	530	525	525	550	545	540	250
	D	28.50	Residue oil.							
(7) 703	A	23.44	60	410	400	400	510	500	495	400
	B	23.13	45	510	500	495	530	523	518	275
	C	25.93	30	530	523	518	550	545	540	250
	D	27.50	Residue oil.							
(8) 704	A	22.48	65	405	400	400	540	545	545	475
	B	22.27	20	540	545	545	550	565	562	200
	C	20.97	25	550	565	562	580	580	568	200
	D	34.28	Residue oil.							
(9) 704	A	21.82	35	450	445	440	540	542	545	375
	B	19.54	25	540	542	545	560	565	560	185
	C	24.63	40	560	565	560	580	570	580	250
	D	34.01	Residue oil.							
(10) 706	A	21.29	55	380	375	370	430	423	430	400
	B	23.07	35	430	423	430	480	475	465	225
	C	25.71	25	480	475	465	515	500	490	350
	D	29.93	Residue oil.							
(11) 707	A	19.76	60	390	380	380	515	515	510	425
	B	18.42	30	515	515	510	520	530	525	200
	C	20.15	25	520	530	525	545	545	540	225
	D	41.67	Residue oil.							
(12) 700, Unused U. S. S. *Arkansas.*	A	24.00	55	390	385	385	480	475	470	325
	B	24.96	30	480	475	470	510	510	505	250
	C	27.57	35	510	510	505	550	550	540	300
	D	23.47	Residue oil.							
(13) 700, Used, filtered U. S. S. *Arkansas.*	A	17.44	45	385	380	380	470	475	468	350
	B	18.40	30	470	475	468	500	503	490	250
	C	17.66	25	500	503	490	535	530	516	215
	D	20.44	35	535	530	516	560	545	530	300
	E	26.06	Residue oil.							
(14) 705, Unused U. S. S. *Arkansas.*	A	21.95	45	425	420	420	545	545	535	450
	B	24.23	40	545	545	535	560	562	552	250
	C	22.83	40	560	562	552	585	575	562	275
	D	30.99	Residue oil.							
(15) 705, Used, filtered U. S. S. *Arkansas.*	A	21.20	45	415	400	390	540	540	525	400
	B	22.32	20	540	540	525	565	568	550	300
	C	23.28	25	565	568	550	585	585	560	300
	D	33.20	Residue oil.							

The above oils were fractionally distilled, giving the following results:

Run No. 2 was a duplication of Run No. 1.

Run No. 4 was a duplication of Run No. 3.

Run No. 7 was a sample of No. 703 oil which had been fractioned as in No. 6; the fractions were then reunited and the recombined oil was refractioned.

Run No. 9 was a run similar to Run No. 7 but using No. 704 oil.

Run No. 12 was a sample of unused No. 700 oil from U. S. S. *Arkansas.*

Run No. 13 was a sample of used filtered No. 700 oil from U. S. S. *Arkansas,* which had been in the system 100 hours.

Run No. 14 was a sample of unused No. 705 from U. S. S. *Arkansas.*

Run No. 15 was a sample of used filtered No. 705 oil from U. S. S. *Arkansas,* which had been in the system 100 hours.

PHYSICAL PROPERTIES OF THE OIL FRACTIONS.

After the distillation, all of the oil fractions had a slightly acrid odor, with the exception of Fraction A, No. 706. This fraction of No. 706 had a strong kerosene odor, which did not disappear on long standing in the laboratory. The slightly acrid odor of the other fractions of oils disappeared in a few days.

The first fractions distilled from the various lubricating oils were very mobile and of a light yellow color, very similar to a "spindle" oil. There was a gradual increase in viscosity and depth of color with increase of the distilling temperature of the fractions of an oil; and in all cases the residue oil had a dark appearance, much resembling a dark cylinder oil. In no residue oil fraction was there any solid matter deposited.

The viscosity, flash point, fire point and specific gravity of the original oil and its fractions were determined by the Oils Laboratory under the supervision of Lieutenant J. L. Kauffman. Viscosity was determined at 100 F. in a Saybolt universal viscosimeter; flash and fire points in the Pensky-Martin closed-

cup apparatus; and specific gravity was determined by means of a pycnometer at 15.6 centigrade. For convenience of reference the item "per cent. of total" is repeated in the following tabulation. The name and number of the oil, the name of the fraction and the explanations on pages 470 and 471, apply to the following oils and fractions of oils:

Name and number of oil.	Name of fraction.	Per cent. of total.	Viscosity at 100 F.	Flash point, degs. F.	Fire point, degs. F.	Specific gravity.
(1) 700	Original oil.	...	199.0	312	364	.937
	A	17.38	64.8	273	300	.917
	B	22.32	108.5	328	362	.930
	C	20.10	202.4	360	408	.937
	D	22.78	428.5	408	458	.942
	E	17.42	2,581.0	404	440	.949
(2) 700, duplicated	A	19.14	64.8	266	310	.9179
	B	19.70	101.8	287	345	.929
	C	18.34	177.3	348	400	.936
	D	19.00	380.0	374	408	.9415
	E	23.82	1,726.0	407	440	.9466
(3) 701	Original oil.	...	205.0	395	455	.887
	A	18.31	99.6	326	360	.8767
	B	18.38	128.0	390	414	.8803
	C	18.49	164.0	388	405	.8827
	D	19.64	250.3	388	412	.8911
	E	25.18	619.8	453	486	.9005
(4) 701, duplicated	A	23.68	102.6	338	364	.878
	B	43.06	165.5	400	435	.884
	C	33.26	511.9	448	475	.898
(5) 702	Original oil.	...	275.0	354	392	.933
	A	27.23	91.2	316	338	.925
	B	24.17	190.0	372	398	.930
	C	28.39	414.0	390	405	.935
	D	20.21	2,358.0	404	446	.943
(6) 703	Original oil.	...	316.0	348	384	.925
	A	24.39	98.1	305	328	.915
	B	22.30	197.2	358	383	.922
	C	24.81	445.0	392	430	.927
	D	28.50	1,402.0	438	450	.933
(7) 703	Fractioned.‡					
	A	23.44	85.7	310	344	.915
	B	23.13	183 8	360	414	.921
	C	25.93	448.0	386	406	.932
	D	27.50	1,557.0	406	470	.9324

‡ Fractions combined and then refractioned.

Name and number of oil.	Name of fraction.	Per cent. of total.	Viscosity at 100 F.	Flash point, degs. F.	Fire point, degs. F.	Specific gravity.
(8) 704	Original oil.	...	255.0	410	436	.904
	A	22.48	126.2	338	362	.896
	B	22.27	179.2	407	435	.900
	C	20.97	248.0	427	442	.904
	D	34.28	603.0	444	468	.9138
(9) 704	Fractioned.‡					
	A	21.82	123.2	358	420	.8965
	B	19.54	171.8	384	438	.8977
	C	24.63	236.2	426	455	.9043
	D	34.01	641.0	438	488	.9131
(10) 706, Ice machine	Original oil.	...	105.0	278	318	.929
	A	21.29	51.3	252	274	.9078
	B	23.07	70.0	278	312	.9190
	C	25.71	160.6	348	364	.9280
	D	29.93	516.5	368	398	.9353
(11) 707, Ice machine	Original oil.	...	128.0	373	401	.872
	A	19.76	73.6	316	342	.8659
	B	18.42	94.2	362	394	.8695
	C	20.15	168.5	372	404	.8732
	D	41.67	222.3	407	438	.8832
(12) 700, Unused, from U. S. S. *Arkansas.*	Oil as received.	...	211.0	319	355	.937
	A	24.00	70.0	280	300	.919
	B	24.96	142.0	330	355	.9325
	C	27.57	303.0	368	416	.939
	D	23.47	1,837.0	414	450	.947
(13) 700, Used, filtered U. S. S. *Arkansas.*	Oil as received.	...	186.0	326	370	.9111
	A	17.44	71.5	288	346	.912
	B	18.40	112.3	326	372	.913
	C	17.66	163.0	360	398	.910
	D	20.44	229.8	371	420	.906
	E	26.06	547.0	417	490	.910
(14) 705, Unused, from U. S. S. *Arkansas.*	Oil as received.	...	178.0	403	464	.883
	A	21.95	108.0	350	390	.878
	B	24.23	140.2	368	405	.877
	C	22.83	185.0	374	405	.880
	D	30.99	380.0	444	468	.892
(15) 705, Used, filtered U. S. S. *Arkansas.*	Oil as received.	...	186.6	410	440	.888
	A	21.20	101.3	326	368	.885
	B	22.32	142.3	362	430	.881
	C	23.28	179.6	410	462	.882
	D	33.20	390.2	455	500	.894

‡ Fractions combined and then refractioned.

DISCUSSION.

The Reduction of the Temperature of Vaporization of Lubricating Oil by the Use of Superheated Steam.

For the purpose of ascertaining the reduction of the vaporizing point of oil by the use of superheated steam, No. 702 oil from the same sample as used in the above experiment was distilled by heat alone. The results as compared to distillation with superheated steam were:

	Dry heat distillation.	Superheated steam distillation.
Oil started to distil	690 F.	380 F.
52 per cent. of the oil distilled over at	740	535
80 per cent. distilled over at	.....	550

When 52.0 per cent. of the oil was distilled by dry-heat distillation, the heat became so intense that the distillation was stopped for fear of melting the glass flask. The distilled oil from the dry distillation had a burned odor, which persisted.

Recombination of Oil Fractions to Produce the Original Oil.

If the fractionation of a lubricating oil as performed in this investigation did not decompose the oil it should be possible to recombine the fractions and produce an oil which had the same properties as the original oil. This was done with No. 701, Run No. 4, and No. 703, Run No. 6. The fractions were proportionately recombined to form a lubricating oil, whose properties compared to the original were:

	Viscosity at—		Flash point, ° F.	Fire point, ° F.	Specific gravity.
	100 F.	130 F.			
No. 701, from oils laboratory	205.0	107.0	395	455	0.887
701, recombined from fractions No. 4	207.2	107.0	374	404	0.887
703, from oils laboratory	316.0	137.0	348	384	0.9251
703, recombined from fractions No. 6	302.0	128.8	336	378	0.927

All the conditions for the distillation of the oil were made as nearly perfect as possible to avoid decomposition or "crack-

ing" of the oil. The large quantity of steam used in the distillation swept the oil vapors rapidly from the still, and prevented excessive condensation and overheating of the oil. The temperature of the steam and oil vapors leaving the still averaged only 10 to 15 F. lower than the temperature of the oil in the still. This was a remarkably good condition to avoid decomposition or "cracking" of the oil. If, in distilling, the oil is allowed to become very hot and considerable condensed oil vapors are allowed to drop back into the hot oil, the heavy-oil residues are broken up into lighter-gravity oils; and carbon is deposited in the still. In no case was carbon deposited in the residue oils of this investigation. It is also to be noted from the data on pages 8, 9 and 10, that the average specific gravity of the fractions is equal to the specific gravity of the original oil. If "cracking" or decomposition took place during the distillation hydrocarbon gases would be evolved, and the average specific gravity of the oil fractions would be much lower than the specific gravity of the original oil. Allowing for a slight loss of oil in wetting the apparatus for distillation and condensation, the volume and weight of the fractions equalled the volume and weight of the original oil.

Results Obtained in Twice Fractioning the Same Oil.

When lubricating oils are fractioned as described in this investigation, then recombined to form the original oil and again fractioned, the second series of fractions have the same physical properties as the first series of fractions of the oil, as shown by oils, Run Nos. 6 and 7, 8, and 9, data on pages 6, 7 and 9. By referring to Plates A and C it is at once apparent that like percentages of oil give the same points in the curves.

The Method Gives Concordant Results.

The method of fractional distillation described above would be of no value if concordant results could not be obtained with the same oil. Concordant results can be obtained with the same oil, as shown by oils, Runs No. 1 and No. 2, and No. 3

and No. 4, data on pages 471 and 473. By referring to graphic charts, Plates A and C, it is shown that like percentages of oil coincide; therefore, the method gives concordant results with the same oil.

Time Required to Operate Test.

The time required for fractioning a lubricating oil is not excessive, as shown by data on page 471. The total time required to heat the water in the boiler, the oil, etc., until the end of the distillation should not exceed four hours.

Safety of Operation.

The method is free from danger to the operator for the reasons that no excessive pressures are used, and the glass flask of the still is completely enveloped in a cast-iron cylinder, which is sufficiently large to completely hold the hot oil if the flask should break.

Number and Size of Oil Fractions.

The sample of oil taken for distillation measured about 600 c. c. When this is fractioned into five fractions approximately 120 c. c. is allowed each fraction. Each fraction is more than sufficient for the careful determination of viscosity, flash and fire points, and specific gravity. These tests are generally recognized as the most important tests of lubricating oils. By dividing the original oil into four fractions additional tests to the above could be made which would in no way affect the accuracy of the result.

On plate A will be found in graphical form the results obtained by plotting average viscosity of the oil fractions against fractions of oil, per cent. Average viscosity was obtained as follows: If an A fraction was 30 per cent. of the total weight of the oil taken for distillation and had a certain viscosity, the point was plotted on the 15 per cent. line, instead of on the 30 per cent. line, because the average viscosity would be the mean

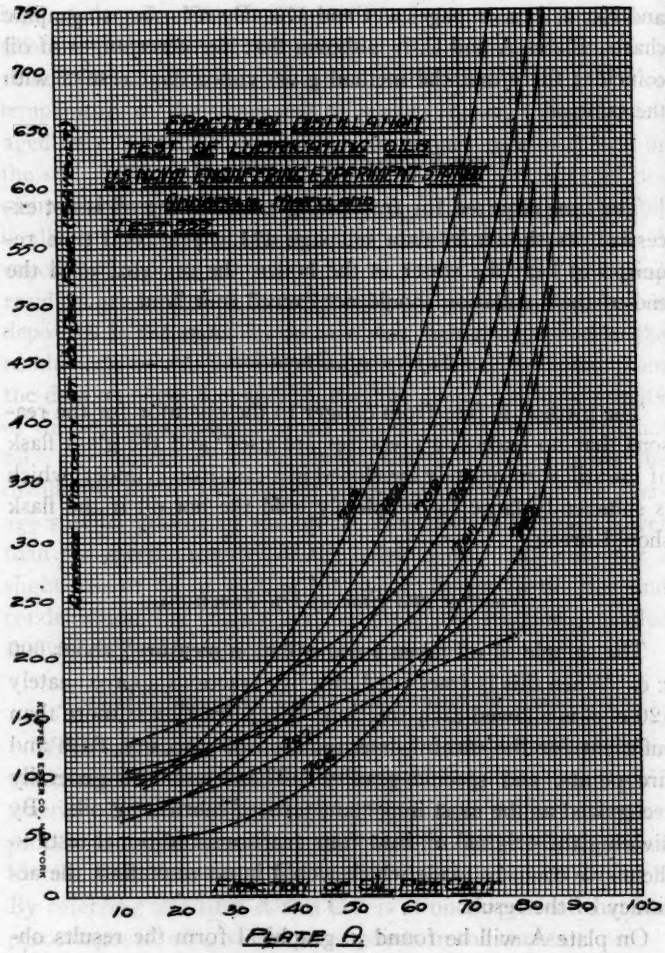


PLATE A.

viscosity of the sample while it was growing from 0 per cent. to 30 per cent. The point was proven experimentally by combining equal volumes of known viscosity A and B fractions and determining the viscosity of the mixture. The mixture A and B fractions could be considered as an A fraction. The viscosity found did not come above the total percentage of A

and B, but it did give the average viscosity of A and B, or a point at $\frac{\text{A plus B}}{2}$ per cent.

Instead of a lubricating oil being composed of oils all having the same viscosity, Plate A shows us that lubricating oil is composed of a large number of oils having a wide range of viscosity, and it also shows us that the viscosity of an oil, as it is known to the trade, is simply the average of the viscosities of all the component oils of a lubricating oil. From Plate A we read, for example, that No. 700 oil contains 40.9 per cent. of oil with a viscosity below 200 sec. Saybolt and 59.1 per cent. of oil with a viscosity above 200 sec. Saybolt at 100 F. From this Plate the following tabulation might be drawn:

	Percentage composition by weight		
Name of oil.	Vis. below 140 sec.	Vis. 140 to 500 sec.	Above 500 sec.
No. 703	25.0 per ct.	39.5 per ct.	35.5 per ct.
702	29.0	43.0	28.0
700	38.5	38.5	23.0
704	17.0	63.0	20.0
701	30.5	54.5	15.0
705	36.0	74.0	0.0

Considering the purpose for which the above oils are intended, it is extremely doubtful if a single argument could be brought forward to justify the presence of such a large percentage of oils having a viscosity below 140 sec. and above 500 sec. Saybolt. These very light and very heavy oils are unsuitable for lubrication, and it is not logical to say that the mixing of two unsuitable oils will make a good lubricating oil.

It is not the object of this investigation to pass judgment on the quality of the above oils, but rather to indicate which component oils of a lubricating oil should be in greatest abundance. The light oils of a viscosity below 140 sec. Saybolt are undesirable because in lubricating machinery they are exposed to conditions of heat, steam, and agitation which rapidly vaporize them, and consequently alter the viscosity of the lubricating oil and shorten the life of the oil. The heavy oils of a vis-

cosity above 500 sec. Saybolt are undesirable because they are more liable to decomposition than the oils of lighter viscosity, producing in decomposition tarry or carbon deposits, causing engine troubles. These heavy oils offset the light oils, and give a viscosity of a medium lubricating oil. When mixed with a light oil these heavy oils do not possess the properties of a heavy lubricating oil—that is, great molecular cohesion; but they temporarily possess the properties of a medium lubricating oil, differing from a good medium lubricating oil in that they change rapidly in viscosity.

From preceding data it is apparent that the life of a lubricating oil in service can be relatively estimated. Lubricating oils containing a large percentage of component oils with a viscosity of 140 to 500 sec. Saybolt at 100 F. should give much longer service than lubricating oils low in these component oils.

On Plate B will be found in graphic form the results obtained by plotting average viscosity of used and unused oils against fractions of oil, per cent. by weight. The oils tested were No. 700 and No. 705, obtained from U. S. S. *Arkansas.* The used oils had been in the system 100 hours. From page 479 we read that No. 700 oil contains 38.5 per cent. of component oils with a viscosity of 140 to 500 sec. Saybolt. Oil No. 705 contains 74.0 per cent. of component oils with a viscosity of 140 to 500 sec. Saybolt. In view of foregoing statements, oil No. 705 should be more stable or less changeable in viscosity than oil No. 700. This is true, as shown from the data and from Plate B.

Attention is especially called to this Plate for the reason that the changes in the two oils of the used and unused are opposite in direction. Oil No. 705 changes with use according to what we would expect from the literature on lubricating oils, *i. e.*, grow heavier in viscosity with use; Oil No. 700 reverses and grows lighter with use.

The decrease in viscosity and specific gravity of No. 700 oil, especially the decrease in gravity of the heavy ends (see data,

page 474, Run 13), is taken as an indication of the decomposition or "cracking" of the used oil.

On Plate C will be found in graphical form the results obtained by plotting the average viscosity of the oil fractions of paraffine and asphalt base oils (data on pages 471 and 473) against the average temperature of volatilization of the oil

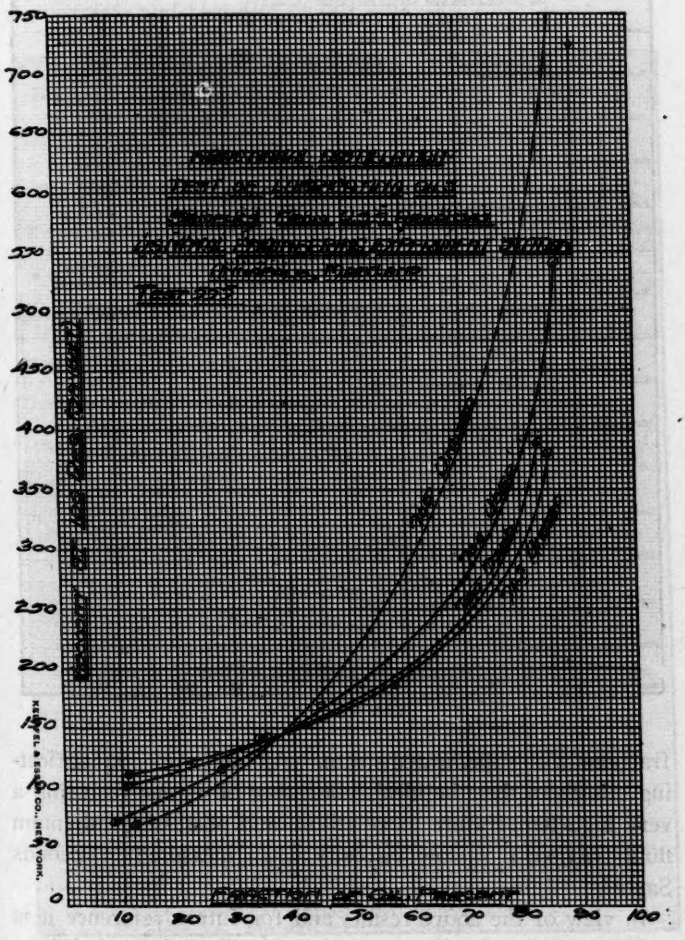


PLATE B

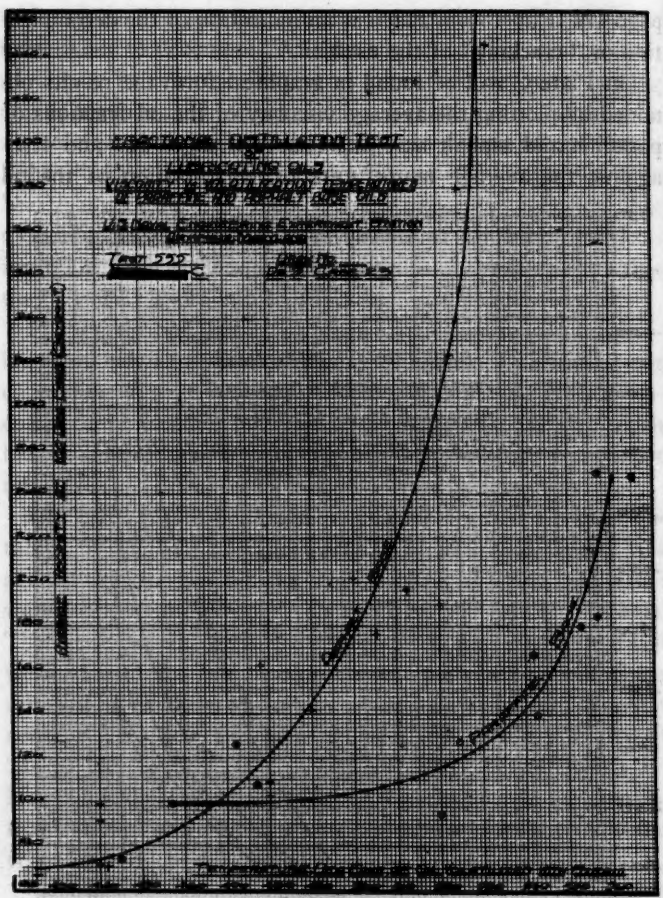


Plate C.

fractions with superheated steam. All authorities on lubricating oils claim that the ideal lubricating oil is one requiring a very high temperature to volatilize, and possessing minimum fluid friction, *i. e.*, low viscosity when measured in seconds Saybolt.

In view of the above results and for future reference it is to be noted that for the same viscosity, Saybolt, asphalt-base

oils are far more volatile than paraffine-base oils. If lubricating oils were not made up of a conglomerate of oils with varying temperature of vaporization and viscosity, as they are at the present time, but were composed of component oils of approximately the same viscosity and with the same temperature of vaporization, there would be a definite relation between the viscosity and volatility of oils.

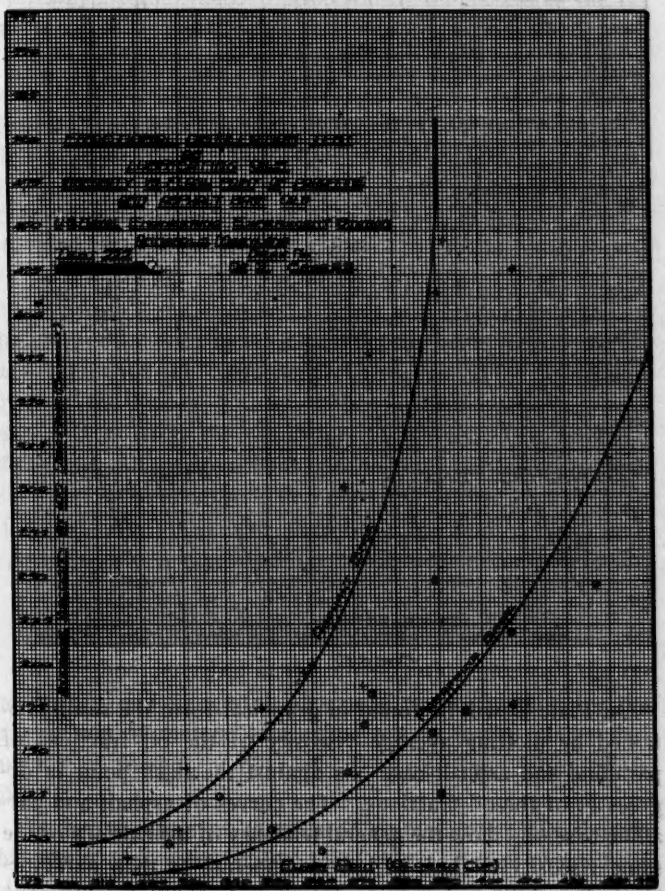


PLATE 2

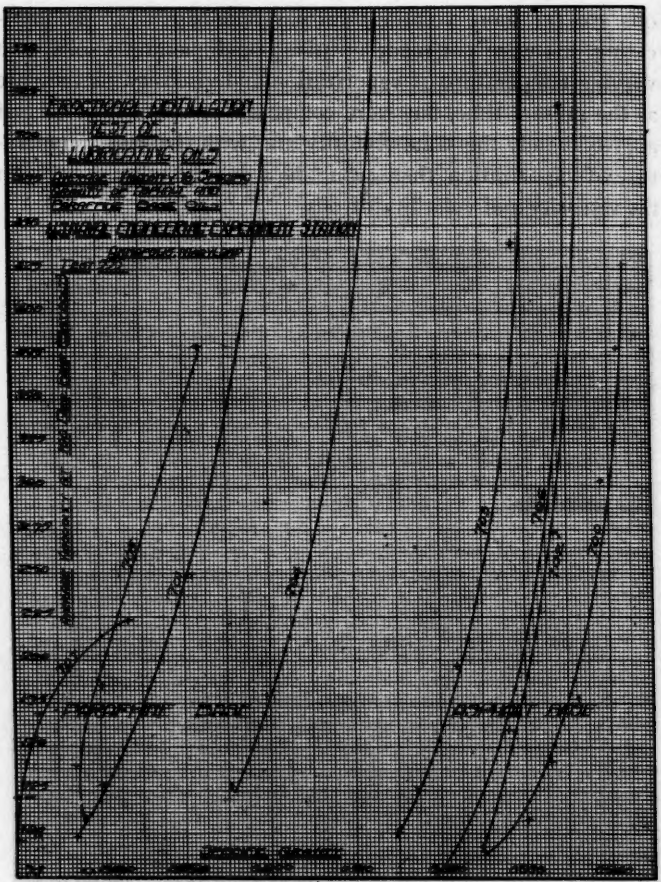


PLATE E

On Plate D will be found in graphical form the results obtained by plotting average viscosity of the oil fractions of paraffine and asphalt-base oils against the flash point in degrees F. of the oil fractions. The curves of asphalt and paraffine-base oils show the general tendency of asphalt oils of the same viscosity to flash at lower temperatures than paraffine-base oils of like viscosity. From data on pages 473, 474

attention is invited to the fact that, when a lubricating oil is fractioned, the first fractions distilled have a much lower flash and fire point than the original oil; in other words, by mixing heavy oils with light ends, the flash and fire points of the mixture are considerably raised above the flash and fire points of the lightest component oil. By subjecting a lubricating oil to fractionation as described in this investigation, information is obtained as to the choice of a lubricating oil possessing the requisite viscosity and having a maximum flash and fire test.

On Plate E will be found in graphical form the results obtained by plotting average viscosity of the oil fractions against specific gravity of the oil fractions. It is shown that for each individual oil there is an increase in specific gravity for proportional increase in viscosity. This point is important when considering used oils; for instance, the loss in specific gravity of "used" No. 700 (oil No. 13, page 474) might be accounted for as the loss by vaporization of light oils of high specific gravity, were it not known that there is a gradual increase of specific gravity with increase of viscosity and temperature of vaporization of the component oils of No. 700. Moreover, knowing the specific gravity of the oil fractions of the "unused" oil, and from the loss in specific gravity of the fractions of the used oil, we are able at once to point out the component oils which have undergone decomposition in use. In the case of "used" No. 700 it was shown that the high-viscosity and high-temperature vaporizing oils underwent decomposition.

CONCLUSIONS.

The method described in this investigation meets all the requirements of a successful laboratory routine method for the examination of mineral lubricating oils as to the quality and quantity of component light oils, heavy oils and intermediate oils.

The investigation of samples of representative oils showed that there was considerable room for the improvement of lubricating oils by the elimination of light and heavy oils, and

by the compounding of mineral lubricating oils from mineral oils having approximately the same viscosity, flash and fire points, specific gravity and temperature of vaporization.

Marked difference between asphalt and paraffine-base oils in regard to temperature of volatilization, viscosity, flash and fire points, and specific gravity, were noted.

Since all lubricating oils, when treated with chemicals, are either unacted on or give reactions not generally understood, it would seem highly important that the herein-described method of fractional distillation be developed to its fullest extent; for the vaporizing temperature of an oil is one of the most valuable means of identifying an oil; especially applicable is this method to lubricating oils, for it increases considerably the use and value of standard tests already familiar to all of us.

From the mechanical viewpoint, provided the oil is well refined, we are only interested in the viscosity of a lubricating oil (latest researches show that the viscosity of the oil controls the coefficient of friction) and the maintaining of this viscosity without change when in use. It is well known to the service what the viscosity of the lubricating oils should be for the different purposes intended, but heretofore there were no known methods which would indicate between equally well-refined oils which oils would longest retain their viscosity without change when subjected to severe usage. The knowledge that shows a lubricating oil will retain its viscosity without change when subjected to severe usage is then equally as valuable, if not more so, than the knowledge of the original viscosity. The maintaining of the viscosity without change when in service is dependent on the following factors:

(*v*) Mineral lubricating oils should be compounded from oils which vaporize within narrow limits.

(*w*) Mineral lubricating oils should be compounded from oils having relatively the same viscosity, specific gravity and flash and fire points.

SUBMARINE ENGINES OF THE GERMAN NAVY.

By Lieutenant C. W. Nimitz, U. S. N., Member.

The modern submarine is usually driven by twin screws, and for surface propulsion depends on internal-combustion engines, one or more on each shaft. The earliest power-propelled submarines were equipped with reciprocating steam engines and coal-burning boilers. The advent of the gasoline engine, with its manifest advantages over the steam plant marked a new stage in the development of the submarine. The Diesel engine, coming immediately after the gasoline engine, soon outstripped the latter as the favorite mode of surface propulsion. A few kerosene engines were built for some of the earlier German submarines, but their use was soon discontinued.

The Germans were quick to grasp the advantages of the Diesel engine and promptly took the lead in its manufacture for all purposes. The scarcity of fuels of all kinds within the boundaries of the German Empire acted as a natural stimulus to the development of economical machinery. The greatest advance was made in the manufacture of machinery for industrial purposes, although considerable progress was also made in the development of marine engines.

The advantages of the Diesel engine over the gasoline and kerosene engines being so apparent, the Germans early attacked that most difficult of Diesel engineering problems—the development of a reliable submarine engine. With a wealth of experience in the general construction of Diesel machinery, it was only natural that the Germans should take the lead in the construction of submarine engines.

The German Admiralty promptly recognized the advantages

within its grasp and lent every encouragement to the manufacturers of the country. What is more important, the Admiralty sent officers and men of the Navy to grow up with the engines and to acquire that knowledge of material so essential to the efficient and reliable operation of light, high-speed reciprocating machinery.

The reliability of the surface motive power is by far the most important factor in the construction of a submarine. No matter how well the hull be designed and arranged; no matter how efficient the electrical equipment or the torpedoes, the submarine will be a failure if her main motive power is not absolutely reliable. Indeed, some builders have discovered, to their sorrow, that the submarine should be built around a reliable engine, instead of designing an engine to suit the hull.

The surface motive power is also the weakest link in the submarine. The rapid increase in size and tonnage has made the problem of power supply more difficult. Though surface speed in a submarine does not assume such strategic importance as high speed in a dreadnaught, the demands for speed made by various admiralties have been partly responsible for the many failures of the engines, and hence the submarines.

The endurance of the submarine can be measured, first, by the endurance of the main engines; second, by the fuel and lubricating-oil supply; third, by the storage battery water supply; and, then, by the endurance of the crew. With engines running properly there is little to do but stand watch, eat and sleep, on a submarine cruising or lying idle on station. On the other hand, if there be engine trouble, requiring constant repair work, the endurance of the crew will be extremely uncertain and limited.

Larger and longer hulls, with increased tonnage, are necessary to increase the speed and the radius of action of the submarine. The reliability of the main engines in the larger hulls can be improved somewhat over the small boat machinery, provided that in the endeavor to obtain the higher powers

necessary there be no reduction in machinery weights per brake horsepower. The habitability of the submarine, which under the very best conditions is poor when compared with surface craft, is also increased with the larger hulls.

Whatever success has been attained by the Germans with their submarines lies more with their more thorough knowledge and experience with Diesel motors than with their larger hulls and tonnage. The endurance of their boats can better be reckoned by the endurance of the engines than by the endurance of the personnel. As long as the engines stay together the crew can remain active and ready for duty. It has been the disheartening struggle with unreliable main motive power that has been the real cause of our apparent submarine failures.

The submarine engines of the German Navy are manufactured by both the Augsburg and the Nuremberg branches of the M. A. N. Co. and by Krupps at Kiel. The Augsburg branch of the M. A. N. Co. manufactures four-cycle engines, while Krupps and the Nuremberg branch of the M. A. N. build two-cycle engines.

At the outbreak of the war the Augsburg works and Krupps enjoyed a monopoly of the submarine engine business for the German Navy, while the Nuremberg works were building a few engines for foreign submarines.

Although the German Admiralty did not look with favor on the Nuremberg type for submarine use, the outbreak of the war found several engines of this type for foreign governments under test or under construction in the Nuremberg works. These engines were promptly installed in German hulls.

The determining feature in the length of time necessary for building a submarine is the length of time required to build the engines. It is difficult to conceive that submarines can be completely built in less than six months' time.

If, however, submarine engines of one type and power are ordered in sufficient numbers, the work can be standardized

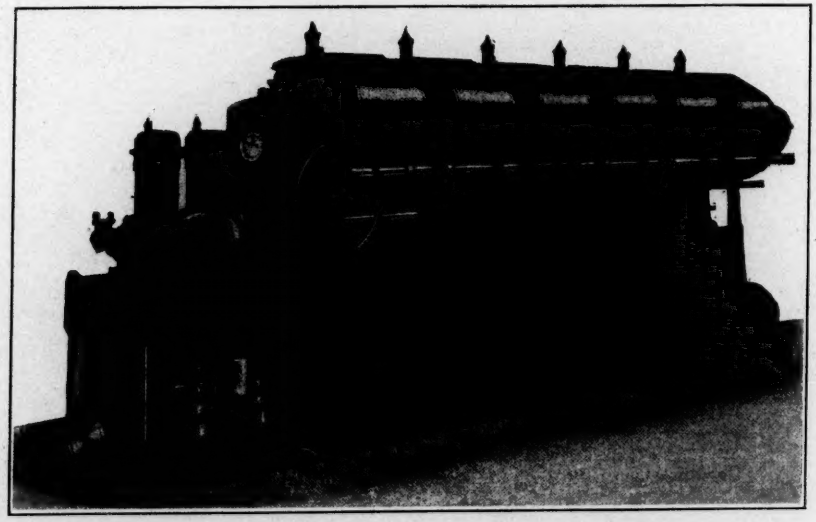
and the time of construction considerably reduced. Such orders are usually not forthcoming, as each class of submarines generally represents an improvement, both in size and power, over the previous class. The number of boats or engines usually allotted by our Government to one contractor in the past has not been sufficient to permit of much reduction in building time by standardization.

THE AUGSBURG TYPE OF SUBMARINE ENGINE.

This engine, at the outbreak of the war, was of the four-cycle, single-acting type, directly reversible by compressed air, and was built to develop 1,000 B.H.P. in six cylinders at 450 r.p.m. The cylinder bore was 310 millimeters and the stroke was 420 millimeters. The engine was directly reversible, using compressed air for the purpose, by moving the camshaft along its length, there being two cams for each valve. Two two-stage air compressors were driven directly off the crankshaft at the forward end of the engine. The bedplate and the crank case were separate steel castings, the bottom of the bedplate being closed by sheet iron. The crank case was box-shaped, with inspection doors abreast each crank pin. The cylinders consisted of cast-steel jackets with cast-iron liners, while the heads were of cast iron. Each head had two fuel valves, one inlet valve and one exhaust valve, one air starting valve and one safety valve. In addition to the safety valve in the head there was another safety valve in the side of each cylinder. All valves were fitted in removable cages or bodies and the exhaust-valve cage and valve were water-cooled. The inlet-valve cages were connected to a common air drum which had a connection to the crank case. The pistons were of cast iron and were oil cooled. Forced lubrication was used on all the main, crankpin and wristpin bearings, the latter being of white metal.

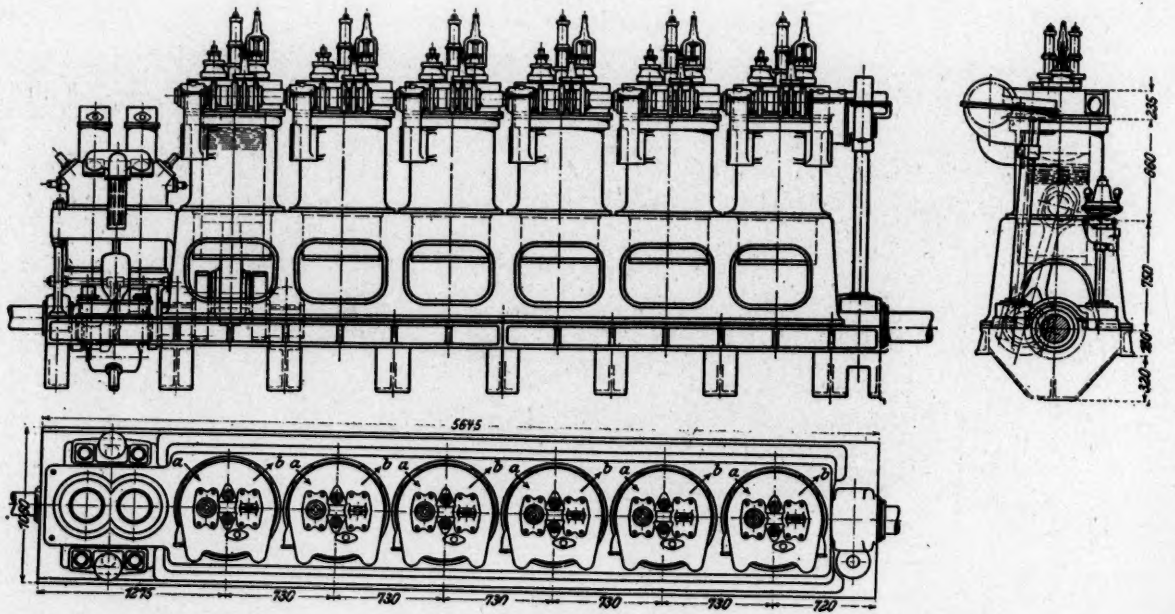
The camshaft was driven by spiral gears and vertical shaft from the after end of the engine.

Two plunger pumps for lubricating oil and piston cooling

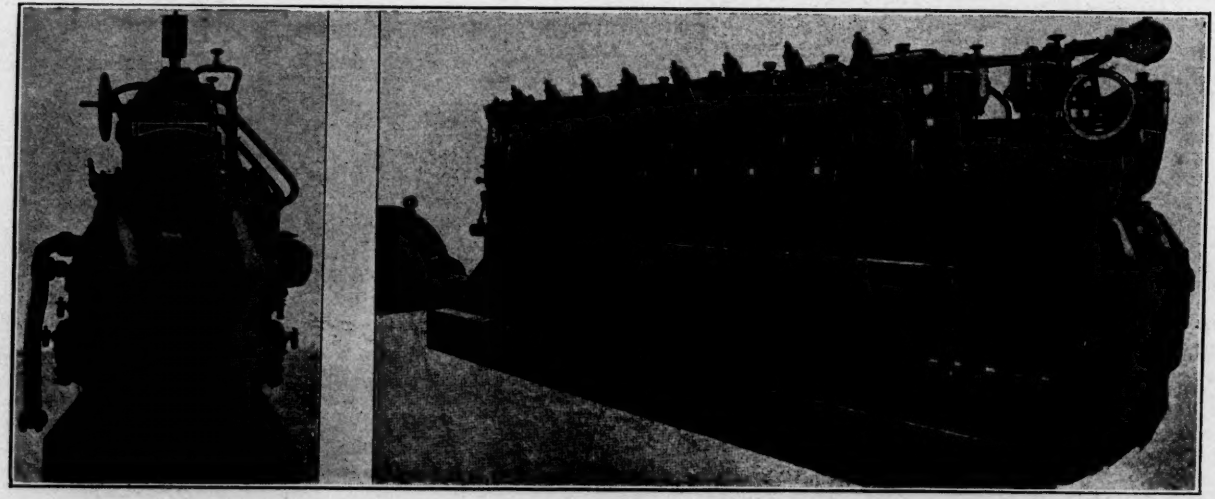


AUGSBURG TYPE OF SUBMARINE ENGINE.

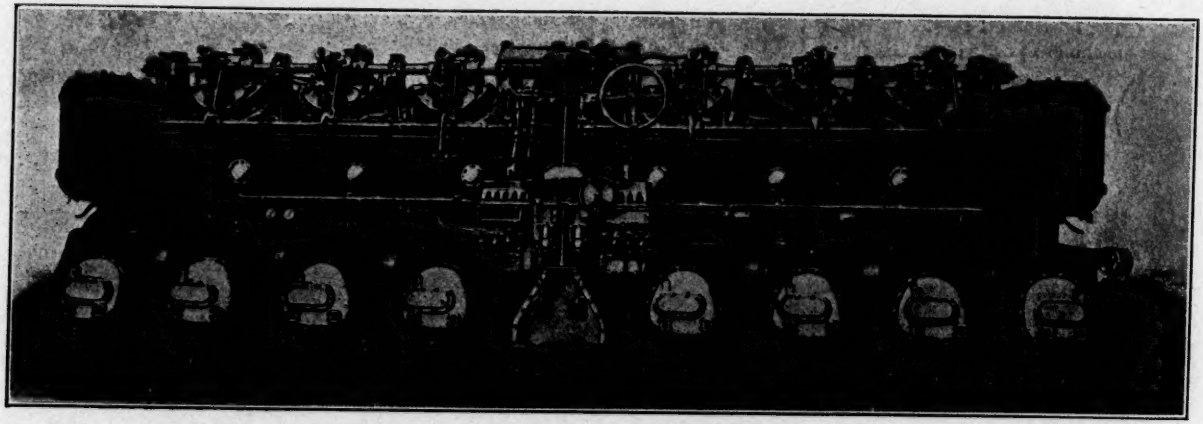
Maßstab 1 : 40.



AUGSBURG SUBMARINE ENGINE.



NUREMBERG TYPE OF SUBMARINE ENGINE.



KRUPP TYPE OF TWO-CYCLE ENGINE.

and two plunger pumps for cooling water, were driven off the forward end of the main crankshaft.

The controlling gear was entirely mechanical and by hand and consisted of four levers; one for starting, one for reversing, one for fuel-pump control and one for controlling the lift of the fuel needle valves. The latter was necessary to economize spray-air pressure and to insure sufficient spray air at reduced speeds.

The reversing-control lever raises the valve levers off the cams, slides the camshaft horizontally until the other set of cams are in position, then lowers valve levers. During reverse the spray air is automatically cut off. This engine had made six full-power duration runs of twenty-four hours each at the time of the writer's visit to the works. It could be throttled down to 70 r.p.m. At full load its fuel consumption was about 200 grams per B.H.P. hour, this being equivalent to .44 pound per B.H.P. hour.

Running at full load the injection-air pressure was about 800 pounds per square inch, the lubrication and piston-cooling pressure 45 pounds per square inch, and the cooling-water pressure about 12 pounds per square inch. The weight of the engine was 51 pounds per B.H.P.

THE NUREMBERG TYPE OF SUBMARINE ENGINE.

Prior to the outbreak of the war the Nuremberg works had under construction or test a number of eight-cylinder, single-acting, two-cycle submarine engines for foreign governments. These engines were rated at 850 B.H.P. at 450 r.p.m., but powers as high as 1,050 B.H.P. have been obtained.

The German engines have eight working cylinders, with scavenger cylinders underneath, the piston being of the stepped type. The cylinder bore is 310 millimeters, the scavenger cylinder bore is 475 millimeters and the stroke of the engine is 340 millimeters. There are two two-stage compressors directly connected to the crankshaft at the forward end of the engine. The compressor dimensions are as follows: bore,

low-stage cylinder, 300 millimeters; high-stage cylinder, 100 millimeters. The high-stage cylinders are mounted in tandem over the low-stage cylinders and the stroke of the combination is 250 millimeters. This engine weighs about 46 pounds per brake horsepower.

The bedplate is of cast steel and is of the open type. A sheet-iron casing, fitted on its lower side, forms the crankpit to catch the oil from the piston-cooling and forced-lubrication system.

The main bearing brasses are cored for water cooling. The main bearing caps are of cast steel, heavily ribbed on top for strength and stiffness.

The engine housing is of cast bronze and is provided with inspection doors on each side in the wake of the crankpins. The housing also contains the scavenger cylinders, the housing being bored to receive the cast-iron scavenger liners from below. The liners are bolted to a horizontal diaphragm running the length of the housing. This diaphragm, with the scavenger jacket, forms an air passage in the upper part of the housing. The inboard halves of the air space for each scavenger cylinder are connected to an induction main running the length of the scavenger cylinders. Air is supplied to this main, either from the boat or through a connection to a ventilator. The outboard sides of the air passages receive the scavenger air from the scavenger cylinders and deliver same to a scavenger receiver running the length of the working cylinders, and just under the exhaust pipe. The suction and discharge valves of the scavenger cylinders are flat spring-loaded disc valves.

The main working cylinders are secured to the tops of the engine housing and are cored for water cooling, the jacket and liner forming one casting. The exhaust belt is in the lower part of the casting and conveys the gas to the outboard side of the engine into a water-jacketed exhaust header running the length of the engine.

The cylinder heads are of iron and are cast separately from

the cylinders and carry all the valves except the relief valve, which is let into the side of the cylinder. The heads are cored for water cooling and carry a fuel valve, two scavenger valves and an air-starting valve.

The pistons are of cast iron and are made in two parts. The lower part forms the scavenger piston and the guide portion of the working piston. The upper part is cored out for oil cooling and is bolted to the lower section. The upper part, or working piston, has its bottom closed by a removable plate. The cooling oil passes from the main bearings into the crankshaft, then through the crankpin and connecting rod to the wrist pin which is in the scavenger piston. From the after end of the wrist pin it passes through cored passages into the working piston. From the top of the chamber in the working piston the oil flows back to the crankpit, via cored passages in scavenger piston and telescopic pipes. The working piston is dished on top and carries five cast-iron rings. The scavenger piston is only about four inches deep and carries two rings. Cast-steel babbitted shoes are bolted on the athwartship sides of the scavenger-piston casting below the scavenger piston. These shoes take the wear due to thrust on the walls of the scavenger cylinder and can be adjusted for clearance by means of liners between the cast-steel shoes and scavenger-piston casting.

A casing of aluminum in the scavenger piston above the wrist pin prevents the spattering of lubricating oil into the hot upper part of the casting. Splash plates are also fitted in the crankpit to prevent excessive splashing of oil on scavenger-cylinder walls.

The cam shaft is directly over the center line of the cylinder heads and it is driven by spiral gears and a vertical shaft at after inboard side of the engine. The shaft carries one cam each for the fuel valve and the two scavenger valves, and two cams for the air-starting valve, one for ahead and one for astern starting.

The vertical shaft contains a jaw coupling with a 30 degree

slip in the jaws. This permits the camshaft on reversal to lap 30 degrees behind the crankshaft, thus placing all cams in proper position for running in the opposite direction.

The air-compressor cylinders and pistons are of cast iron and the pistons are of the tandem type.

Forward of the compressors are mounted eight fuel-measuring pumps, one for each cylinder. They are driven by a rocker arm and eccentric from the main shaft.

The handling gear is mounted on the forward end of the engine and consists in a control wheel which automatically controls the reversing and the speed of the engine.

The crankshaft drives at its forward end a cooling-water pump, a lubricating-oil pump and a fuel-oil supply pump.

This type of two-cycle engine, with scavenger cylinders underneath the working cylinders, possesses many advantages for submarine use when it is properly constructed. Its main disadvantage lies in the increased height of engine, while its many advantages are as follows:

First: The wrist pin, being located in the scavenger piston, can be made much larger for bearing surface and can be kept out of the hot part of the cylinder.

Second: The engine will be shorter in length and narrower in breadth than one whereon the scavenger pumps are driven off the end or back of the engine.

Third: The larger number of scavenger pumps make for greater reliability, and a cylinder breakdown will not totally disable the engine, as might be the case with an engine having just one or two scavenger pumps.

It is, of course, more difficult to remove a piston from this type of engine, but when properly built the necessity of removing pistons is reduced.

KRUPP TYPE OF TWO-CYCLE ENGINE.

This engine, as built just prior to the war, was a six-cylinder, single-acting, two-cycle engine, directly reversible by compressed air, and developed about 900 brake horsepower at 450

r.p.m. At higher speeds of revolution about 950 brake horsepower was attained.

The weight of this engine is about 45 pounds per brake horsepower.

This engine is built almost entirely in bronze, the cylinders and pistons being of cast iron, and the crankshaft, connecting rods, wrist pins and valves being of steel.

At each end of the engine a double-acting scavenger pump is driven directly off the crankshaft. In the middle of the engine and driven directly off the crankshaft are two two-stage air compressors. This arrangement makes a very long engine. The handling station is in the middle of the engine. The camshaft is on the inboard side of the engine and carries two cams for each of the following valves in the head: one fuel valve, one air-starting valve and three scavenger valves. The three scavenger valves are operated by one lever from the camshaft. The operation of reversing consists in moving the camshaft horizontally until the second set of cams is in position. During this shift the rollers are held off the cams. The scavenger pumps deliver to a receiver back of the engine. Two connections from each receiver supply the scavenger valves in the heads. The fuel and air-starting valves are inclined toward the inboard side of the engine in passing through the cylinder heads.

It is quite possible, and also probable, that the power of the Augsburg four-cycle and the Krupp two-cycle engines has been increased by adding two more working cylinders. In the case of the Augsburg engine there would then be available at least 1,350 brake horsepower on each shaft of a submarine. The Krupp engine would probably increase its power to the same amount, by adding two more working cylinders, but in this case the engine would be excessively long.

Increasing the power by a slight increase of engine dimensions would not involve great changes in the outfit of patterns. It is quite possible that the Krupps have increased their power

by slightly increasing their engine dimensions, rather than by adding more cylinders.

There is no single company as yet in this country which builds its own hulls and manufactures its own engines. The manufacturers of our engines in all cases purchase their crankshafts either finished or rough turned, and some even buy their castings. Such conditions do not make for rapid construction and will partly account for the length of time required to build a submarine. The present state of the steel, iron and copper markets should be a warning to our own country of how wartime demands for material for all forms of new construction work may seriously embarrass the builders of submarines and submarine engines.

Within the last two years several American firms have undertaken the manufacture of high-speed marine Diesel engines suitable for submarine work, and while it may take some time to obtain a thoroughly reliable submarine engine, it will be forthcoming, provided these manufacturers receive proper encouragement in the shape of orders from the Government.

It is to be noted with satisfaction that the Navy Department has finally decided to construct submarines and engines in some of the navy yards. Such a course, even though under a license, where all of our design work is done for us, cannot fail to increase our material knowledge. By these means and by a more thorough and lengthy course of instruction of the personnel in the propulsive machinery, we can increase the efficiency of our submarines.

The crews of submarines are necessarily very small and the loss of one member from the engineer's force usually means a reduction of about 10 per cent. in that force, while on a destroyer the loss of one man means about a 2 per cent. reduction. On a battleship this loss causes only about ½ per cent. reduction of the engineer's force. The submarine service should have first choice in the matter of keeping the complements full, or even overflowing. The submarine service

should also be made so attractive in the matter of higher pay that there will be a constant waiting list of applicants.

In the matter of personnel the Germans are very fortunate, for not only has Germany a larger proportion of skilled mechanics in her population than we in ours, but she has in addition a compulsory system, whereas we must depend on volunteers for our recruits.

It is obvious that under a compulsory system there would never be shortages in our crews, nor would we have difficulty in obtaining skilled mechanics or operators as machinists in our submarines. Highly skilled men forced to serve would naturally choose to serve in submarines where their skill and patriotic zeal is put to its hardest test.

In our volunteer system we have occasionally received fresh from civil life recruits highly skilled in operating large internal-construction machinery; such cases are, however, extremely rare. It has been a case of training and instruction from the first day on the submarine until the expiration of enlistment. Fortunately for our service, a large proportion of our submarine men reënlist. If it were not for that, we would indeed be struggling vainly in our efforts to produce efficient and mobile submarines.

U. S. S. *JACOB JONES.*

DESCRIPTION.

By W. F. Sicard, Associate.

The *Jacob Jones*, Torpedo-Boat Destroyer No. 61, is one of six destroyers, of the same class, authorized by an Act of Congress approved March 4, 1913. These vessels were designated as Torpedo-Boat Destroyers Nos. 57 to 62, inclusive.

The circular calling for bids for these vessels was issued May 20, 1913. Plans and specifications were issued to prospective bidders May 31, 1913, bids were opened at the Navy Department, at noon, August 4, 1913, and the vessel was delivered to the Government February 10, 1916.

The contract for the construction of the *Jacob Jones* was awarded to the New York Shipbuilding Company, of Camden, N. J., at that company's bid of $825,000, and dated October 15, 1913. Of the contract price $486,000 was allotted to machinery. The contract contained the usual provisions embodied in the act entitled "An Act to increase the naval establishment," approved August 3, 1886. The vessel was to be completed within 24 months of the date of contract and there were the usual penalties for under speed and over time.

The *Jacob Jones* is a twin-screw vessel propelled by Parsons turbines and designed for a speed of 29½ knots, at a displacement of 1,150 tons, with the turbines developing 18,625 shaft horsepower. There is a cruising turbine and reduction gear, connected to the port shaft, for propelling the vessel when cruising. Steam is supplied by four Thornycroft boilers.

PRINCIPAL HULL DIMENSIONS.

Length between perpendiculars, feet and inches	310-00
on L.W.L., feet and inches	310-00
overall, feet and inches	315-03
Beam, extreme, feet and inches	30-06½
on L.W.L., feet and inches	29-09
Ratio of length to beam (L.W.L.)	10.391
Draught to L.W.L., mean, feet and inches	9-04½
Displacement corresponding, tons	1,090
Coefficient of fineness, block	.439
midship section	.714
load water plane	.625

GENERAL DESCRIPTION.

The *Jacob Jones* is of the usual raised-forecastle type. There are two masts, fitted with wireless, signal yards, etc., and four smokepipes.

Forecastle Deck.—This deck includes the pilot house and provides for the stowage of anchors and bow gun. On top of the pilot house, and extending to the vessel's sides, is the bridge, fitted with a steering wheel. Above the bridge is a searchlight platform.

Main Deck.—The main deck is continuous from stem to stern. The officers' quarters, galley, radio room, etc., are forward. Abaft the forecastle is a weather deck, on which is located the battery, and there is a deck house aft, containing the crew's toilet rooms, etc. On top of this deck house is a steering platform and a searchlight.

Platform Deck.—The platform deck extends forward and aft of the machinery spaces only. It contains the crew's quarters and petty officers' quarters and the steering gear.

Hold.—In the hold are located trimming tanks, magazines and stores. The chain lockers are forward, and the machinery spaces occupy the entire mid-portion of the hold. The reserve-feed tanks are located in the after boiler compartment and the fresh-water tanks in the forward boiler compartment, on both port and starboard sides. The engineer's store rooms are located on the starboard side of forward and port side of after boiler compartments.

BATTERY.

The battery consists of four 4-inch rapid-fire guns, one on the forecastle, two on the main deck forward, on port and starboard sides at the break of the forecastle, and one aft, on the main-deck center line, abaft the after deck house.

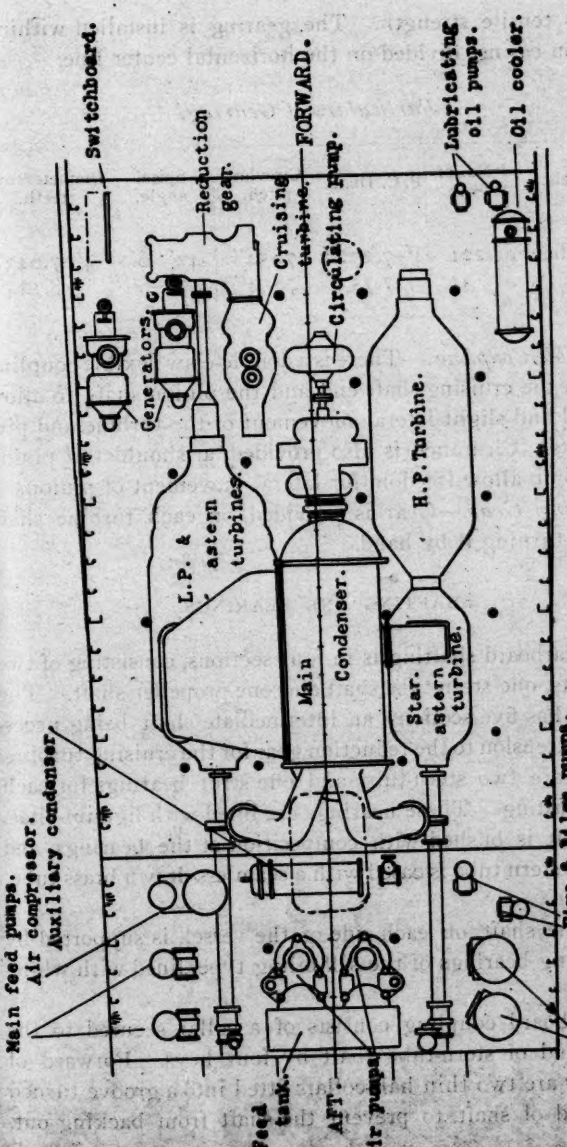
The torpedo equipment consists of four 21-inch twin torpedo tubes mounted on the main deck, two port and two starboard.

MAIN PROPELLING MACHINERY.

The main propelling machinery consists of Parsons turbines in combination with a geared cruising turbine. There are two shafts. On the starboard shaft there is one high-pressure turbine, abaft which there is an independent backing turbine. On the port shaft there is a low-pressure turbine, and a backing turbine within the low-pressure casing. Forward of the low-pressure turbine is a geared cruising turbine. The arrangement is shown in Plate I. The cruising turbine is arranged to uncouple, through the medium of a clutch coupling, from the turbine shaft when revolutions greater than about 2,500 per minute are desired, these revolutions corresponding to a speed of 20 to 25 knots, and approximately 1,600 shaft horsepower. The turbine glands are steam packed and formed of knife-edge strips and gland rings. There is a steam connection to the space between the knife-edges and the gland rings. The turbine steam chests are packed with dummy rings, forming labyrinth packing, to prevent leakage.

REDUCTION GEAR.

The gearing is of the double-helical type, divided in two, with right-and-left-hand teeth, having an angle of 42 degrees 6 minutes. The total length of the gear wheel faces is 16 inches. The reduction-gear wheel is of forged steel, Class "B", in halves, in a plane at right angles to the shaft. The pinion and sleeve pinion are of nickel-steel, of about 90,000



U.S.S. JACOB JONES. ENGINE ROOM AND AUXILIARY ROOM.

PLATE I.

pounds tensile strength. The gearing is installed within a cast-iron casing, divided on the horizontal center line.

Particulars of Gearing.

Wheels.	No. of teeth.	P. C. Diam.	Circular pitch.	Spiral angle.	Diameter over teeth.
Main wheel .	221	4′–7.272″	.7854″	42°–6′	4′–7.643″
Pinion . . .	31	7.753″	.7854″	42°–6′	8.124″

Flexible Coupling.—There is a double-claw flexible coupling between the cruising-shaft end and the pinion shaft, to allow for axial and slight lateral movement of the turbine and pinion shafts. Clearance is also provided, at shoulder of pinion bearings, to allow freedom for lateral movement of pinion.

Turning Gear.—Gear is provided, on each turbine shaft line, for turning it by hand.

SHAFTING AND BEARINGS.

The starboard shafting is in four sections, consisting of two line shafts, one stern-tube shaft and one propeller shaft. The port side has five sections, an intermediate shaft being necessary for extension to the reduction gear for the cruising turbine.

There are two stern-tube and one strut bearings for each line of shafting. These bearings are lined with lignum-vitae. Each shaft is bushed with composition at the bearings, and within its stern tube is cased with a seamless-drawn brass tube, shrunk on.

The line shaft, on each side of the vessel, is supported by three spring bearings of the self-oiling type, lined with white-metal.

The inboard coupling consists of a collar secured to the forward end of stern-tube shaft by four keys. Forward of this collar are two thin half-collars fitted into a groove turned in the end of shaft, to prevent the shaft from backing out,

the whole being through-bolted to the after coupling disc on the line shaft.

The outboard couplings are of the ordinary sleeve type, secured to the shafts by four feathers and two cross-cotters.

Clutch Coupling.—There is a clutch coupling operated by a handwheel, bevel gears, worm and rack, for disconnecting the reduction gears from the port main turbine shaft. It is at the after end of the intermediate shaft, the general particulars are as follows:

Outside diameter, inches	15½
Diameter over all, inches	15⅝
Coupling rings, diameter, inches	13½
thickness at flange, inches	02½

This clutch was perfectly satisfactory. During trials the cruising turbines were disconnected without a brake in continuity of trials. Cruising-turbine throttle was closed, clutch run out, and main throttle opened, in each case the reduction in speed was scarcely appreciable. Reversing with cruising turbine in operation is easily accomplished as the clutch is easily thrown out while the shafts are slowing or while they are momentarily stopped.

Particulars of Shafting.

Intermediate shaft:	
Outside diameter, inches	06½
Diameter hole through shaft, inches	03
at journal, inches	06½
Length, feet and inches	7–08 11/16
Line shafts:	
Two sections, outside diameter, inches	09½
Diameter of hole through shaft, inches	06⅛
at journals, inches	09⅛
Total length of each section, starboard, feet and inches, about	39–08 7/16
port, feet and inches	39–11 13/16
Stern-tube shafts:	
Outside diameter, inches	09½
Diameter hole through shaft, inches	06⅛
Composition casings at bearings, thickness, inch	00 7/16
outside diameter, inches	10⅛
between bearings, thickness, inch	00½
Length of shafts, feet and inches	25–09½

Propeller shafts:	
Outside diameter, inches	09½
Diameter hole through shaft, inches	06⅝
reduced at ends to diameter, forward, inches	02⅝
aft, inches	02⅝
Composition casings at bearings, thickness, inch	00 7/16
outside diameter, inches	10⅝
Length of shafts, feet and inches	21–00 1/16
Three line-shaft bearings, length, inches	12
Stern-tube, forward bearing, length, feet and inches	2–04 7/16
after bearing, length, feet and inches	3–06
Strut bearings, length, feet and inches	3–06
Coupling discs, diameter, inches	16
thickness, inches	02
Inboard coupling, diameter of collar outside, inches	16½
inside, inches	10½
length of collar, inches	08½
thickness of half collars, inches	01½
Coupling bolts, number each coupling	8
diameter, taper, at face of coupling, inches	01 1/16
Outboard coupling, length of sleeve, inches	32⅝
diameter of sleeve, outside, inches	12

PROPELLERS.

There are two three-blade propellers of manganese-bronze, the blades cast with the hubs. The starboard propeller is right-hand and the port propeller left-hand. The blades are machined true to pitch and the hub is secured by two keys and a nut. The taper of shaft through propeller is 1 inch in diameter per foot length. The actual particulars are as follows:

	Starboard.	Port.
Diameter, inches	94	94
Pitch, inches	82.875	82 7/16
Helicoidal area, square inches	4,764	4,731
Projected area, square inches	4,284	4,254
$\frac{P}{D}$	0.8816	0.882
Weight, pounds	3,728	3,693

MAIN CONDENSING APPARATUS.

Main Condenser.—There is one main condenser, oval in section, located in the engine room at center line of ship. It

is of the curved-tube type. The tubes are curved in a vertical direction and expanded in both tube sheets. The principal dimensions are as follows:

Diameter of shell, maximum, inside, feet and inches	8–08½
Thickness of shell, inch	00 5/16
tube sheet, inches	01½
Tubes, number	5,472
diameter, inch	00⅝
thickness, mils	49
length between tube sheets, feet and inches	12–00¼
Cooling surface, square feet	10,800
Main exhaust nozzle, area through, square feet	35
Augmenter suction connection, diameter, inches	15
Air-pump suction, diameter, inches	15
Circulating-water inlet, diameter, inches	30
Circulating-water outlets, two, diameter of each, inches	21
Auxiliary exhaust connection, diameter, inches	09
Bleeder connection, diameter, inches	02½

Main Air Pumps.—The condenser is provided with two, Blake, vertical, twin-bucket, single steam cylinder, beam, air pumps, 14 × 28 × 28 × 18 inches. The suction and discharge pipes are 11 inches and 9 inches in diameter, respectively.

Main Circulating Pump and Engine.—The main circulating pump is of the Worthington bi-rotor, volute type, direct connected to a Terry, type C. M. B., 36-inch wheel, steam turbine. The rotors are right and left, 27½ inches diameter, with 8 vanes each. The pump has the following connections:

2 21-inch diameter sea suctions.
1 07-inch bilge suction.
1 30-inch diameter discharge to condenser.
1 05½-inch diameter discharge to augmenter.

Feed and Filter Tank.—There is a feed and filter tank located in the auxiliary room. The capacity of the filter space is 285 gallons and of the tank proper 510 gallons, total 795 gallons. The filter chamber is in the top of the tank and so arranged that the entering water will flow over the top of a

plate and leave in a similar manner, thus assuring the filtering material being always submerged.

ENGINE-ROOM AUXILIARIES.

Pumps.—A list of engine-room pumps and their connections is given in Table I.

Auxiliary Condenser.—The auxiliary condenser, connected through the auxiliary exhaust pipe to all auxiliary machinery and the augmenter, is located in the auxiliary room. It is of the curved-tube type, the tubes being curved in a vertical direction, and rolled into the tube sheets at both ends. The principal dimensions are as follows:

Diameter of shell, maximum, inside, inches	21
Thickness of shell, inch	00.165
Tubes, number	316
diameter, inch	00⅝
thickness, inch	00.165
length between tube sheets, feet and inches	5-10
Cooling surface, square feet	300
Diameter auxiliary exhaust connection, inches	07
air-pump suction, inches	04
circulating-water inlet and outlet, inches	04

Auxiliary Air Pump.—See Table I.

Auxiliary Circulating Pump.—This pump is of the Worthington, 3-inch, double-suction, Class B.S., high-speed, volute, impeller type. For connections see Table I.

Vacuum Augmenter.—There is an augmenter condenser, provided with a S. & K. vacuum augmenter, fitted in connection with the main condenser. It is of the straight-tube type with steam jet and water seal. It is connected to the main condenser and the water seal. The auxiliary exhaust connection was installed as an experiment and very satisfactory results have been obtained. The augmenter uses the auxiliary exhaust steam only and produces a better vacuum than the Parsons augmenter which uses live steam. On the preceding destroyer it was necessary to by-pass considerable auxiliary exhaust steam to the main condenser. With the augmenter as fitted on the *Jacob Jones* it was found unnecessary to by-

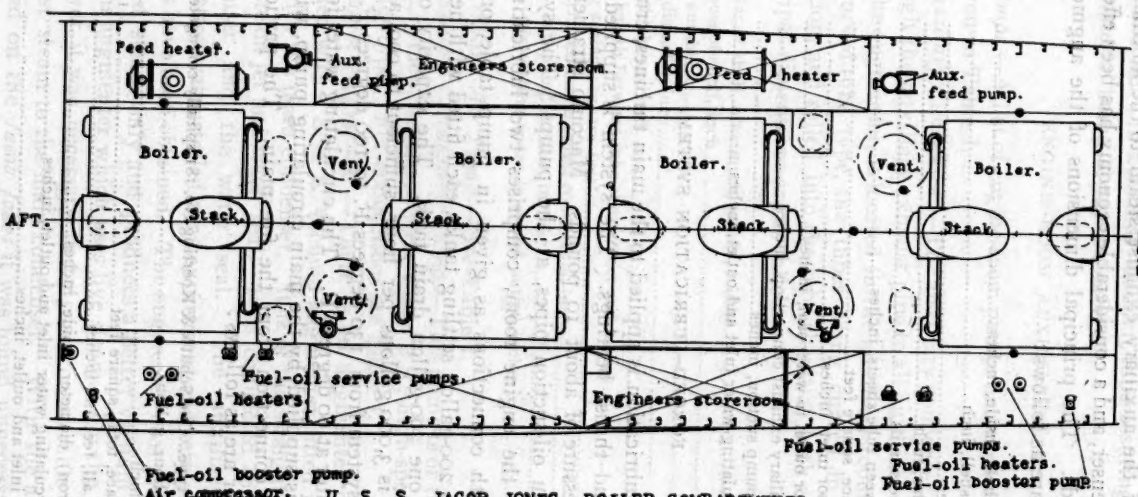
pass any of the auxiliary exhaust steam to the auxiliary or main condenser, and a considerable economy has been effected at all speeds. The principal dimensions of the augmenter condenser are as follows:

Diameter of shell, inside, inches	22½
Thickness of shell, inch	$00\frac{5}{16}$
Tubes, number	392
diameter, inch	$00\frac{5}{8}$
thickness, mils	49
Distance between tube sheets, inches	60
Cooling surface, square feet	320
Diameter vapor inlet, inches	13
vapor outlet to water seal, inches	13
auxiliary exhaust connection, inches	03
air-pump suction, inches	15
circulating-water inlet and outlet, inches	$05\frac{1}{2}$

FORCED–LUBRICATION SYSTEM.

Forced lubrication is applied to the main turbines, cruising turbines and thrust bearings. The system is supplied with oil at a pressure of about 10 pounds. Macomb strainers are installed, in oil-suction pipes, at the pumps. The system, located in the engine room, comprises two lubricating-oil pumps, with connections as given in pump table, one oil cooler, two 200-gallon settling tanks, each fitted with heating coils, and one 200-gallon drain tank. The capacity of the oil cooler is 3,000 gallons per hour, sufficient to cool all the oil in the system from 130 degrees F. to 100 degrees F. with cooling water at 60 degrees F. The circulating water for the oil cooler is supplied by the main circulating pump, with an emergency connection from the fire main. The particulars of oil cooler are as follows:

Number and type	One Schutte & Koerting No. 181 spirally-corrugated film.
Capacity per hour, gallons	3,000
Film space, inch	$00\frac{3}{16}$
Cooling surface, total, square feet	131.76
Length over all, feet and inches	7–07½
Shell (cast iron), diameter, inside, inches	21⅛
Diameter circulating water inlet and outlet, inches	03
oil inlet and outlet, inches	$02\frac{1}{2}$



U. S. S. JACOB JONES. BOILER COMPARTMENTS.

PLATE II.

WATER SERVICE.

There is no independent water-service system, water being supplied from the hose connections at the pumps.

BOILERS.

There are four water-tube oil-burning boilers of the Thornycroft type, arranged in pairs in two separate watertight compartments as shown in Plate II. They are designed for a working pressure of 265 pounds per square inch per gage, and are operated under the closed-fireroom system of forced draft. Each boiler has an independent smoke pipe.

Data for One Boiler.

Working pressure, pounds per square inch, per gage	265
Number of furnaces	1
Furnace volume, cubic feet	620
Area through smoke pipe, square feet	18.5
Smoke pipe area / Furnace volume	0.03
Heating surface, square feet	5,984
Heating surface / Furnace volume	9.65
Oil burners, number	11
Oil burners, type	Bureau air registers. Koerting burners.
External height, including lagging, feet and inches	13–06
width, including lagging, feet and inches	16–02
length, including lagging, feet and inches	14–03½
Tubes, outside diameter, inches	01¼ and 01⅛
thickness, mils	109 and 120
number, 1¼ diameter	2,160
1⅛ diameter	170
Downcomers, inside diameter, inches	09
thickness, inch	00 5/16
number	2
Drum, steam, number	1
inside diameter, inches	42
thickness, shell, inch	00 13/32
tube sheet, inches	01 13/32
Water drums, number	2
inside diameter, inches	19
thickness, shell, inch	00 7/16
tube sheet, inch	00 11/16

Diameter main steam stop valve, inches	07
auxiliary steam stop valve, inches	03½
safety valves (two duplex), each valve, diameter, inches.	04
main and auxiliary feed, stop and check valves, inches...	03
bottom-blow valves, inches	01½
surface-blow valves, inches	01½

The water drums of boilers are welded with no seams.

FUEL—OIL SYSTEM.

Steam is generated by the combustion of fuel oil, burned by the mechanical atomization system.

The fuel-oil pumps, pipes and heaters are located in the firerooms. The system in each fireroom consists of a light-service booster pump, two heavy-pressure duplex, service pumps and two oil heaters in each boiler compartment together with all the necessary fittings, piping and strainers. The pump connections are as given in Table I, and there is, in each fireroom, one Schutte-Koerting 1¼ by 3½-inch triplex, hand oil pump, with two cranks, for supplying oil to the burners when raising steam. There is a Schutte-Koerting 1½-inch interlocking duplex-pressure oil filter. The air registers are of the Bureau of Steam Engineering type with Koerting burners and there is an air chamber in the combined discharge from fuel-oil service pumps.

FIREROOM AUXILIARIES.

Forced-Draft Blowers.—There are two forced-draft blowers in each boiler compartment. The fans are of the Green radial-flow type mounted on vertical shafts in the base of the fireroom ventilators from which the air supply is taken. Each fan is driven by a vertical G. V. type Terry steam turbine, located immediately below the fan. The forced-draft blower data is as follows:

Fan, greatest diameter, inches	43½
least diameter, inches	38½
width, overall, inches	12 9/16
Number of vanes	8
R.p.m., designed, about	1,600

Pumps.—Table I gives the list of fireroom pumps and connections.

Pneumatic System.—There is a torpedo air compressor located in the forward port side of the auxiliary room. There is also a pneumatic system for the use of the engineer's department, consisting of one 11 × 11 × 12-inch Westinghouse air compressor, located in the after starboard corner of the after boiler compartment, with one 12.8 cubic feet capacity air-storage tank. There is a 1-inch diameter copper pneumatic main, extending through the machinery spaces with downtakes, in each compartment, ¾-inch diameter terminating in ¾-inch hose valves.

Feed-Water Heaters.—There is a horizontal feed-water heater in each boiler compartment. Particulars of each feed-water heater are as follows:

Type	Schutte & Koerting, spirally-corrugated film, Navy type.
Capacity	125,000 pounds water per hour, heated from 90° F. to 200° F.
Number of tubes	28
Tubes, outside diameter, inches	02 9/16
thickness, mils	95
Heater, length over all, feet and inches	8–00 7/8
diameter of shell, inches	29½
thickness of shell, inch	00 5/8
Material, shell	Steel plate, Class B.
heads and covers	Cast steel, Class B.
Exhaust connections, inlet, diameter, inches	08
outlet, diameter, inches	06
Feed connections, inlet and outlet, diameter, inches	04

MAIN STEAM PIPES.

A 7-inch diameter main steam pipe leads from the stop valve on each boiler. The pipes from the forward boilers unite into a common pipe, 9½ inches diameter, leading aft on the port side, and those from the after boilers into a similar pipe on the starboard side. Both 9½-inch pipes lead to the engine room and branch to the main turbines and the cruising turbine. There is a valve in each 9½-inch pipe on the engine-room side of the bulkhead. There is a pipe abaft the engine-

TABLE I.—U. S. S. "JACOB JONES."—PUMPS AND PUMP CONNECTIONS.

No.	Pumps.	Size.	Kind.	Size.	Suction pipes from.	Size.	Discharge pipes to.
		inches.		*inches.*		*inches.*	
2	Main air.....	(2) 14 × 28 × 18	Twin, vertical, bucket, S. A.	11	Condenser through water seal.	9	Feed tank.
1	Main circulating.....	Terry C. M. B. 36-inch wheel.	Turbine driven.. centrifugal.	2 of 21	Sea.....	30	Main condenser.
				7	Bilge.....	5½	Augmenter.
						2½	Oil cooler.
2	Main feed	15 × 10 × 16.....	Vertical, piston... D.A., single.	5½	Feed suction. 3″ reserve feed tank.	4	Main feed discharge.
				5½	Channel ways.....		
2	Auxiliary feed.....	15 × 10 × 16.....	Vertical, piston... D.A., single.	5½	Feed suction.....	4	Auxiliary feed.
				3	Reserve feed.....	1½	Hose connection.
				1½	Hose connection.....		
2	Fire and bilge.....	7 × 7 × 12.....	Vertical, piston... D.A., single.	4	Sea.....	2½	Fire main.
				4	Drainage.....	3½	Overboard.
				1½	Hose connection.....	3½	Flushing.
						1½	Hose connection.
1	Auxiliary circulating	3″ B. S. high-speed volute.	Turbine driven, centrifugal.	4	Sea.....	4	Auxiliary condenser.
1	Evaporator feed.....	4½ × 6 × 6.....	Vertical, piston... D.A. single.	2	Sea.....	1½	Evaporators.
				1½	Distiller circulating..		
1	Distiller, fresh water.	3½ × 4 × 4.....	Vertical, piston... D.A., single.	1½	Distiller reservoir tank.	1½	Reserve feed.
						1½	Ship's tanks.
						1½	Cofferdam.
						1½	Main feed tank.
							Washroom, aft.
2	Fire and bilge, F. R.	7 × 7 × 12.....	Vertical, piston... D.A., single.	3½	Drainage.....	3½	Overboard.
				4	Sea.....	2½	Firemain.
				1½	Hose connection.....	1½	Hose connection.
2	Lubricating oil.....	6 × 5½ × 8.....	Vertical, piston... D.A., single.	3	Lubricating-oil tanks	2½	Bearings.
						1½	Settling tanks.
4	Fuel-oil service.....	6 × 3½ × 8.....	Vertical, piston... D.A., duplex.	2½	Oil-storage tanks.....	2	Burners.
				2½	Booster-pump discharge		
2	Fuel-oil booster.....	6 × 5½ × 8.....	Vertical, piston... D.A., single.	3	Oil-storage tanks.....	2½	Service pump supply.
				2½	Deck-hose connection	2½	Oil-storage tanks.
			Light service.....			2½	Deck-hose connection

room main stop valves, cross connecting the main steam pipes on port and starboard sides of the vessel. Maneuvering pipes lead from the valves at the starting platform to the L. P. ahead and to each astern turbine. The main steam pipes are of seamless-drawn steel.

AUXILIARY STEAM PIPES.

There is a 3½-inch diameter stop valve on each boiler for supplying the auxiliary steam line which leads throughout the machinery spaces with connections to the various auxiliaries and with a branch, forward, to the deck machinery, forward heating system and galleys. The auxiliary steam pipes are of seamless-drawn steel.

AUXILIARY EXHAUST PIPES.

An auxiliary exhaust pipe, of copper, extends throughout the machinery spaces and elsewhere and is connected to all of the auxiliary machinery. Connections are provided for direct exhaust into either the main or the auxiliary condenser, the augmenter, the feed-water heaters, or into the atmosphere through the after escape pipe. There are also connections for admitting exhaust steam to the stage of ahead turbines to which the pressure in the auxiliary exhaust pipe corresponds at the various speeds. There is a 7½-inch diameter escape pipe abaft each smoke pipe.

FEED PIPES.

The main and auxiliary feed pipes take their suctions from the feed-suction main, which extends from the bottom of the main feed tank to the auxiliary feed pump in the forward fireroom. The main is 8 inches diameter in the engine room, decreasing in size to 5½ inches in the forward fireroom. The suction branches to all feed pumps are 5½ inches diameter. All feed pumps are arranged to feed any boiler via the feed heaters or by-passes. The feed discharges from pumps are 4 inches diameter and the branches to each boiler 3 inches in diameter. There is a 4-inch independent main connecting

the auxiliary feed pumps. All boiler feed pipes lead to air chambers, on the feed-check valves, at the boilers.

MACHINE TOOLS.

There is, located at forward end of after fireroom, on starboard side, a 12-inch Davis lathe, having a 6-foot bed and provided with all necessary tools and attachments. It is driven by a Reliance motor 1¼ horsepower direct-current, adjustable-speed type with speed ranges from 500 to 1,500 r.p.m.

ELECTRIC PLANT.

The dynamos are located as shown on Plate I. The installation consists of two horizontal, compound-wound, direct-current, 25-kilowatt G. E. generators, each driven by a Curtis steam turbine. Each generator will deliver at normal load 200 ampères of current at 125 volts, when running at 3,600 r.p.m.

HEATING PLANT.

A steam-heating plant, working at a pressure of about 50 pounds, is provided for the living and other spaces. Steam for heating is taken from the auxiliary steam pipe.

TORSION METERS.

Gary-Cummings torsion meters, one for each line shaft, are installed for measuring the shaft horsepower.

EVAPORATING AND DISTILLING APPARATUS.

The distilling apparatus is located in the auxiliary room. There are two evaporators, two distillers, and one evaporator feed-water heater, with all accessories, arranged to operate in single or double effect.

Data for One Evaporator.

Type	Schutte-Koerting, horizontal U-tubes.
Diameter, inside, inches	22
Thickness of shell, inch	$00\frac{7}{8}$
Height overall, feet and inches	5–$06\frac{1}{4}$
Tubes, number	32

Data for One Distiller.

Type	Bureau, vertical, straight tubes.
Diameter, outside, inches	12½
Thickness of shell, cast iron, inch	00 7/16
Length between tube sheets, inches	29
Tubes, number	81
diameter, inch	00⅝
thickness, mils	49
Cooling surface, square feet	32
Diameter circulating-water connections, inches	03
vapor inlet, inches	04

Data for Evaporator Feed-Water Heater.

Type and number	Bureau 1
Number of tubes	8
Inside diameter of tubes, inches	02½
Thickness of tubes, mils	83
Heating surface, tubes, square feet	13.68
tube sheet	03.81
total	16.49
Diameter vapor connection, inches	05½
Area vapor connection, square inches	23.75
through tubes, square inches	39.26
Ratio vapor area to tube area	1 : 1.65

TRIALS.

The standardization trials, were conducted on December 17, 1915.

The official speed and revolution curve gave the following mean revolutions per minute of the propellers as necessary to obtain the speeds stipulated by the trials:

Speed in knots.	R. P. M.
12	184.6
16	244.2
20	308.1
25	416.7
29½	559.8

All trials were successful in all particulars.

TEST OF WARD BOILER.

By Lieutenant S. M. Robinson, U. S. Navy, Member.

A Ward boiler was tested by a Board consisting of Lieutenant Commander H. C. Dinger, U. S. N., Lieutenant S. M. Robinson, U. S. N., Lieutenant A. M. Penn, U. S. N., and Mr. A. Conti.

Particulars of the boiler tested are as follows:

Type of Boiler	Ward Straight Tube.
Total tube heating surface	4,405 square feet.
Volume of furnace	476 cubic feet.
Fuel used	Elk Refining Co., paraffine base.
Number of burners	7
Type of burners	Bureau of Steam Engineering Type I.

DESCRIPTION OF TEST OUTFIT.

The apparatus and arrangement for test are indicated on Plate 1. The boiler was fitted with wooden air box at front, fitted with air lock. Air was supplied by means of a motor-driven Sturtevant blower. As the blower motor was of the constant-speed type, the air supply was regulated by means of a butterfly shutter in blower discharge and a sliding door on front of air box.

Feed Water.—Feed was supplied by means of two horizontal Blake plunger pumps discharging through two Reilly feed-water heaters, the boiler being fed through two check valves. Temperature of feed was taken at the feed valves.

Oil.—Oil was supplied by means of two Blake horizontal, piston, simplex oil pumps, discharging through Reilly oil-fuel heater (coil type) to the fuel-oil main at boiler front; a special air chamber was fitted on the discharge line. There were fitted in addition three large air chambers near oil heater. These additional air chambers were found necessary in order to reduce fluctuations in pressure. Temperature of oil was taken at boiler front. Oil used was Elk Refining Company's paraffine-base West Virginia oil, of the following characteristics:

Special gravity at 60 degrees	.8245
Baumè, degrees F.	40.1
Flash point, degrees F.	144
Fire point, degrees F.	270
Viscosity (Engler), at 60 degrees F.	9.25
Acidity	None.
B.T.U.	19,879
Sulphur	None.

Method of Measuring Water.—Feed was measured by taking account of number of tanks, from tank calibrated by actual water measurement and found to contain 4,875 pounds. Tank measurement was taken at beginning and end of test, and correction applied. Correction was also applied for difference in boiler-water level at beginning and at end of test.

Method of Measuring Oil.—Oil was measured by weighing actual oil every ten-minute interval. After having weighed tank, oil was run into a pump-suction tank. The height of oil was brought up to a mark at the end of each ten minutes.

Steam generated was blown into atmosphere, except that used for pumps and feed heaters.

Gas Analysis.—A Hayes CO_2 apparatus was connected to the base of the stack. Samples of gas were taken every half hour. The results of the gas analysis are believed to be unreliable, as they show very little excess of air used over that theoretically possible; as there was smokeless combustion at all times, this would be impossible. An electric pyrometer was

fitted in base of stack. This was checked during trials with a mercurial pyrometer and found to be accurate.

Quality of Steam.—A throttling calorimeter was fitted just below steam connection to stop valve.

TESTS.

Three tests were run, the first test to determine the efficiency of the boiler when burning oil at the rate of 2,300 pounds of oil per hour, the second when burning oil at the rate of 1,500 pounds of oil per hour, and the third test when burning oil at the rate of 3,100 pounds of oil per hour.

Test No. 1.—Approximate rate, 2,300 pounds of oil per hour. Test begun 5:00 P. M., February 7, 1916, ended 11:00 P. M., February 7, 1916. Seven burners fitted with 5/64-inch tips and 3/64-inch plugs were used.

Two oil pumps, two feed pumps, two feed heaters were in operation throughout the test.

There was some vibration on several occasions during the trial due to lack of care with air regulation.

There was no observable smoke during the entire trial. A small excess of air was supplied.

At 8:00 P. M. one feed pump broke down and operated indifferently during the remainder of run.

The casing of the boiler, especially the left side in wake of lower rows of tubes at back, became red hot in places, indicating insufficient lagging behind the brick.

Test No. 2.—Approximately 1,500 pounds of oil per hour. Test begun midnight, February 7, 1916, ended at 6:00 A. M., February 8, 1916. Seven burners were used. The burner used was 1/16-inch tip and 3/64-inch plug.

One oil pump, one feed pump, one feed heater were in operation.

The conditions were very steady during the entire trial. The combustion was excellent. The casing became quite hot at sides near back of boiler.

There was no observable smoke during the entire run.

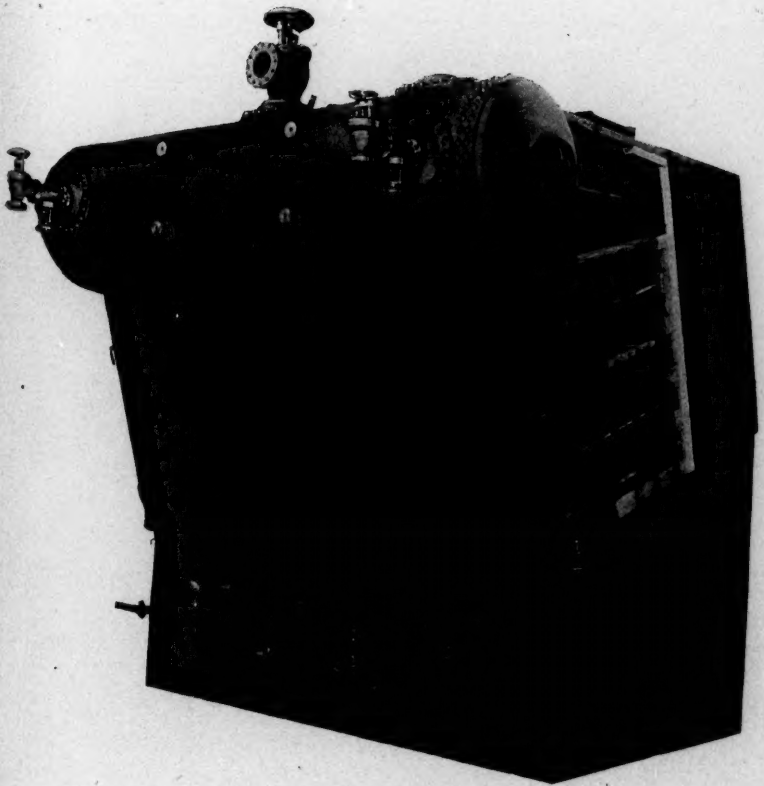


PLATE I.—WARD TEST BOILER.

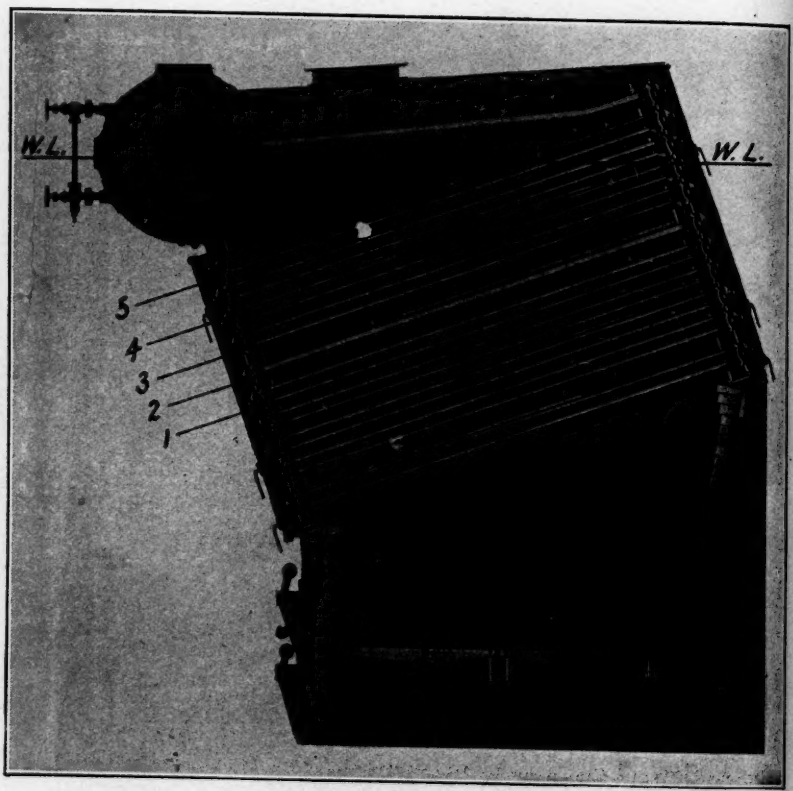
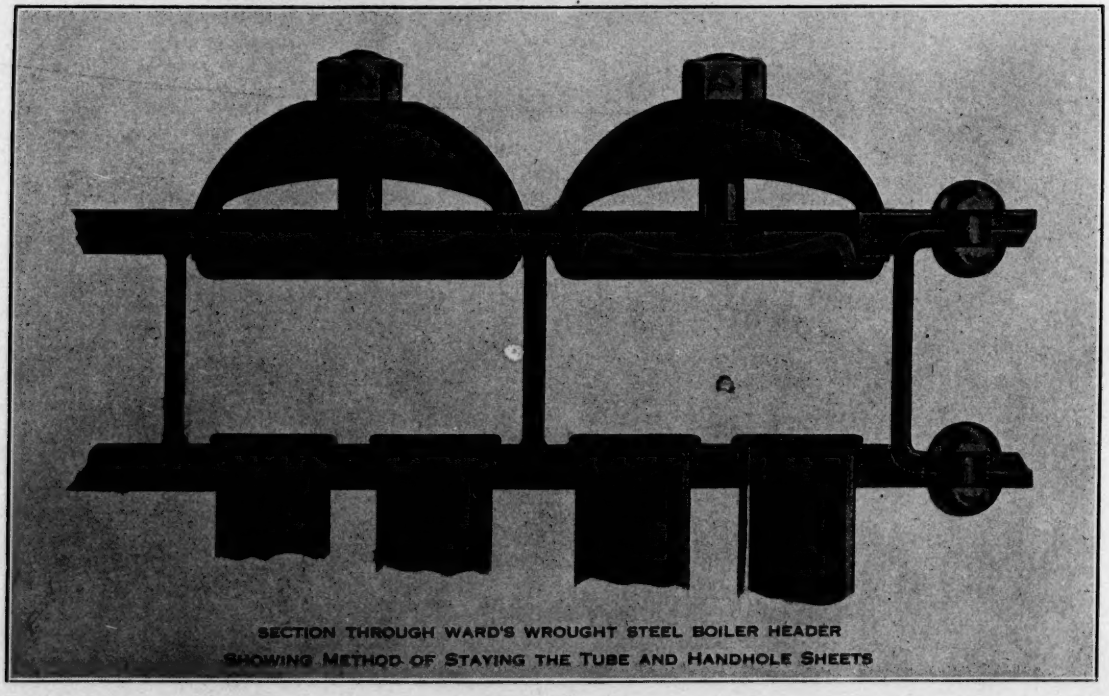


PLATE II.



SECTION THROUGH WARD'S WROUGHT STEEL BOILER HEADER
SHOWING METHOD OF STAYING THE TUBE AND HANDHOLE SHEETS

PLATE III.

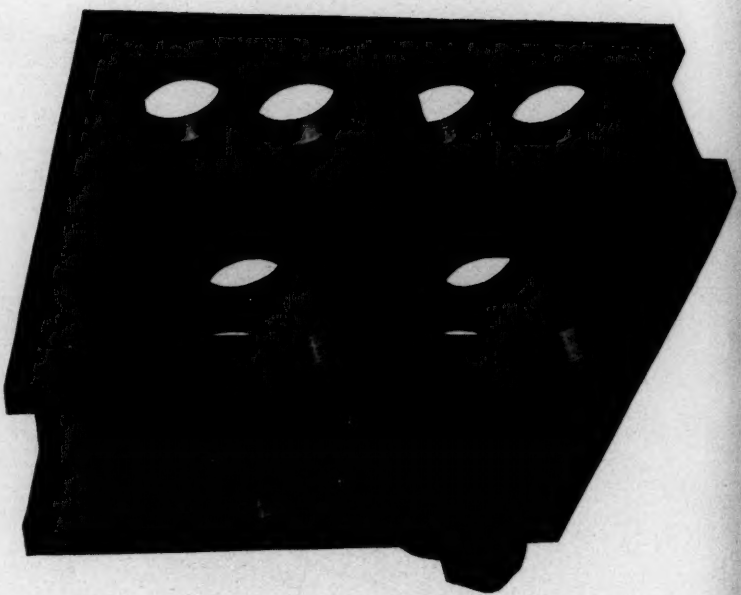


PLATE IV.—SECTION SHOWING HEADER CONSTRUCTION.

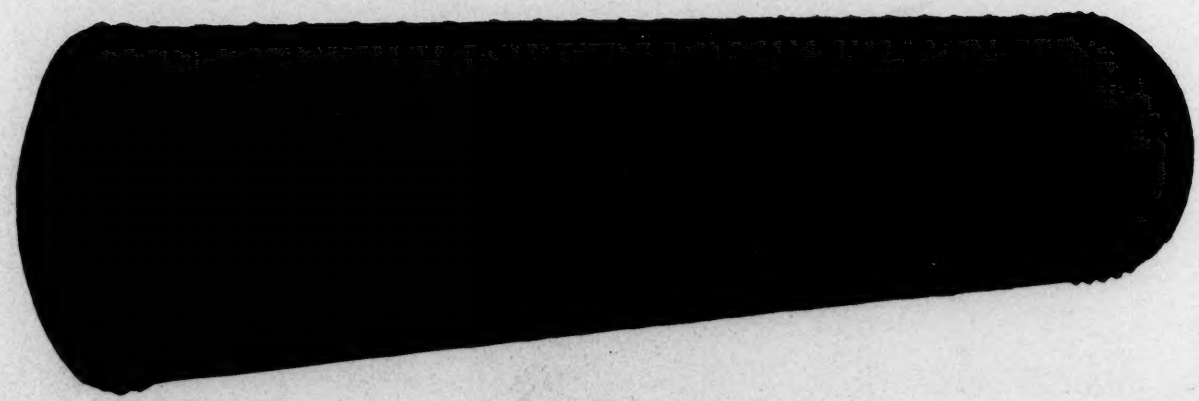


PLATE V.—STEAM DRUM.

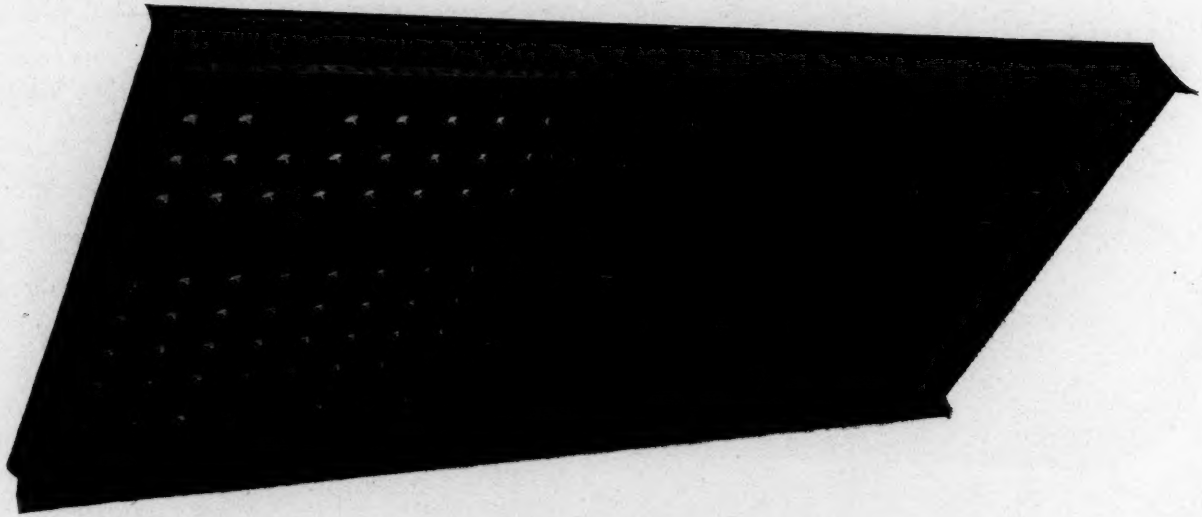


PLATE VI.—FRONT FLITCH, OR HEADER.

PLATE VII.—BACK FLITCH, OR HEADER.

At end of test boiler was let down, brickwork appeared in very good condition; the bottom appeared to be somewhat uneven, indicating that magnesia under bricks was disintegrating to some extent.

During February 8th the feed pump was overhauled. Additional asbestos millboard was placed behind metal side casing, and the brickwork, especially that on left side of boiler, was repaired.

Test No. 3.—Approximate rate 3,100 pounds of oil per hour. Fires were started at 12:30 P. M., February 9, 1916. Data for test was taken from 1:30 P. M. Test ended 6:30 P. M. On account of insufficient time to get boiler warmed up to proper conditions, it was decided to use data for four hours, from 2:30 P. M. to 6:30 P. M. Seven burners were used. The burner used was 3/32-inch tip and 3/64-inch plug.

Two oil pumps, two feed pumps and both feed heaters were used.

Owing to lightness of oil, difficulty in forcing a sufficient amount through the burners was experienced when oil was heated. The oil was therefore not heated after the first hour. With cold oil the required amount of oil was burnt.

There was no observable smoke during the entire run.

The left side of casing became red hot during test, indicating that there is insufficient insulation in casing as built.

Synopsis of Tests.

Test No.	2	1	3
Approximate rate of oil burner per hour required by schedule 7,360	1,500	2,300	3,100
Oil actually burned, pounds per hour	1,509.7	2,440.7	3,082
Pounds of water evaporated from and at 212° F. with oil as fired of 19,879 B.T.U	16.73	16.48	16.21
Reduced to 19,000 B.T.U	15.99	15.75	15.49
Guaranteed for 19,000 B.T.U	15.66	15.33	15.00
More than guaranteed	.33	.42	.49

COMMENTS ON OPERATION OF BOILER.

The boiler steamed freely under all the tests. The circulation is evidently satisfactory. Small propellers were placed in two of the intermediate four-inch tubes, with a rod running through the handhole. The movements of these rods indicates a down-flow in the four-inch tube at all rates of steaming.

After the regular test the Charles Ward Engineering Works, at the suggestion of the Board, installed propellers in several of the two-inch tubes to ascertain the character of the circulation. The result was as follows. (See Plate 2.)

Number 2 began to revolve in about five minutes after the fires were started; the direction of rotation showed down-flow in these tubes. Number 1 hesitated slightly, showing some inclination to turn left-handed, which was probably due to the suction caused by the down-flowing water from Number 2. This condition changed before steam was raised, and Number 1 began to revolve in the opposite direction, showing that the water was flowing up the lower bank of two-inch tubes. Numbers 3 and 4 began to revolve left-handed in about ten minutes after the fires were lighted, the speed increasing until they were running like windmills, so fast that the revolutions could not be counted; the direction of rotation showed down-flow in these tubes. Number 5 remained stationary until there was about 50 pounds of steam on the boiler, and then it commenced to revolve in a left-handed direction, indicating that as soon as the water level raised on the high side of the tubes, due to the steam-making condition, there was a down-flow through the two-inch tube mentioned.

From these experiments it is inferred that the flow is upward through all the two-inch tubes below the middle baffle, and downward through all the two-inch tubes in the upper portion of the boiler, the velocity being quite rapid, for, as stated, the propellers were revolving in the neighborhood of 200 r.p.m. or more. The high velocity of the propellers in the

small tubes is due to the fact that the shafts were of very small diameter and consequently there was very little friction.

To establish the volume of water flowing down through the four-inch tube, one of the propellers was removed and installed in a short piece of pipe (as shown by the accompanying sketch), this being connected to the city water main and the propeller operated under a pressure of about 40 pounds, the discharge being throttled until the propeller made approximately the same number of revolutions as it did in the boiler. Readings from the water meter over a period of five minutes showed that there was about 50 gallons per minute passing through this tube. This, however, is only an indication of the amount of water flowing through the four-inch tubes, as the friction on the propeller was very much greater under the higher pressure conditions existing in the boiler.

The foregoing results are very interesting, and are believed to throw a new light upon the problem of circulation in boilers of this general type of construction. They also indicate that the baffling as arranged secures the benefit of the counter-current principle throughout the boiler.

The casing as constructed is insufficiently insulated for high rates of combustion. This can, however, be easily remedied by providing additional insulation, and, if brickwork is provided with anchor bolts, injury due to panting will be avoided.

No distortion of any of the lower tubes was observed, and it is apparent that there is no difficulty in giving the lower tubes an ample supply of water.

COMMENTS ON DESIGN OF BOILER.

The design gives a very compact straight-tube boiler with good circulation.

The baffling as arranged gives a counter-flow action between water and gases through the boiler, which apparently secures very high efficiency.

Tabulated Results of Tests.

1	Test No.	2	1	3
2	Barometer	30.79	30.73	30.55
3	Length of test averaged, hours	5	6	4
4	Number of burners used	7	7	7
5	Bore of burner tips, inch	$00\frac{1}{16}$	$00\frac{5}{64}$	$00\frac{3}{32}$
	Steam Observations.			
6	Steam pressure, pounds gage	194.75	196	197.6
7	Calorimeter shell pressure, ins. of Hg	2.82	2.8	2.6
8	Calorimeter shell temperature, degrees F.	305.3	305.7	304.6
9	Feed temperature of boiler, °F	183.6	113	168.3
10	Water actually evaporated, lbs. per hour	23,532	35,082	45,878
	Oil Observations.			
11	Oil burned, pounds per hour	1,509.7	2,440.7	3,082
12	Oil pressure at burners, lbs. gage	258	255	283
13	Oil temperature at burners, °F	163	141	50.5
	Air Observations.			
14	Fireroom pressure, ins. of water	2.34	4.91	4.96
15	Fireroom temperature, degs. F	62	74	64
16	Stack temperature	418	475	520
17	Smoke, Ringelmann scale	0.1	0.1	0.1
	Flue Gas Analysis.			
18	Carbon dioxide	12.48	12.61	13.14
19	Oxygen	3.08	2.97	1.89
20	Carbon monoxide	.....	.....	.....
21	Nitrogen (by difference)	84.44	84.42	84.97
	Calculated Results.			
22	Moisture in steam, fraction of 1 per cent	.616	.606	.69
23	Factor of evaporation	1.0738	1.1466	1.089
24	Pounds of water evaporated per pound of oil	15.59	14.37	14.89
25	Pounds of water evaporated from and at 212° F. per lb. of oil:			
26	for oil of 19,379 B.t.u.	16.73	16.48	16.21
	reduced to 19,500 B.t.u	16.41	16.17	15.90
27	Water evaporated per square foot of H.S. per hour, actual	5.342	7.964	10.415
28	Water evaporated per square foot of H.S. per hour, from and at 212 degrees F	5.736	9.131	11.34
29	Pounds of oil burned per square foot of H.S. per hour	.3427	.5541	.6996
30	Pounds of oil burned per cubic foot of furnace per hour	3.17	5.125	6.475
31	Boiler efficiency, per cent	81.68	80.45	79.13

The manner of staying the water chambers is considered efficient, and no trouble should be expected from the design of water chambers.

There is ample space in the header beneath the edges of the lower tubes to allow for the collection of sediment, should this be contained in the feed water.

ANALYSIS OF STRESSES IN CONNECTING RODS.

By Karl F. Smith, Lieut., J. G., U. S. N., Member.

In all machine design at the present day there seems to be a tendency on the part of the producer to base his design upon certain empirical formulae which practice has "shown to be satisfactory," without examining too closely the rationality of the formula itself. While in most cases such a procedure is justifiable, there arrives a time when changing conditions may warrant a change in our formulations.

The object of this paper is to show in particular that the stresses deduced by the formulae for connecting rods now in general use are misleading and give results much below the actual stress in the rod. While this is undoubtedly true, there is little danger of failure due to the fact that the allowed fiber stress is purposely kept low in these parts. However, in the design of high-speed engines, where weight is a big factor in design, the danger from these excess stresses becomes more pronounced.

The fact that one stress which, contrary to general opinion, is of extreme importance has generally been neglected except for the arbitrary "factor of safety," will be pointed out, and a formulation which is rational for all rods and for any material shown.

Before starting our formulation we must first examine what conditions exist in a connecting rod at any instant, that is with the rod at rest, and later see what effect the rotation will have on these stresses.

We immediately see that at rest this rod may be either a compression or tension member—compression on the down

stroke and tension on the up stroke. It is seen immediately from the relation

$$\text{Stress intensity} = \frac{\text{Load on rod}}{\text{Area of rod}'} = \frac{P}{A'},$$

that the strength in tension is amply sufficient for any probable load on the rod. We have, therefore, only to investigate the failure under compression.

Due to the form of the wrist and crank-pin bearings we see also that this rod partakes of a dual nature; that is, in the plane parallel to the shaft the rod may be considered as a column with "fixed ends," while in the plane of oscillation it is a column with "pin ends," and free to take up any angular position in this plane.

All theories and experiments show that the collapsing load is much greater for fixed ends, and we have only to consider collapse due to loading, which may cause *bending in the plane of oscillation.*

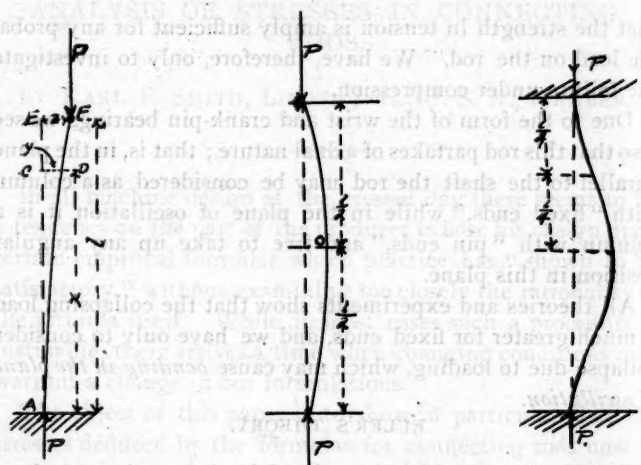
EULER'S THEORY.

Euler's theory, which is rational for long columns where the effect of direct compressive stress is negligible, gives values for failing loads which agree fairly well with experimental data *within certain limits.*

It has been shown experimentally that in perfectly straight, long columns of homogeneous material, with a compressive load axially applied, the column will buckle and collapse under a much lighter load than is required to produce *failure by crushing in a short column of the same cross-section area and form.* This failure is caused by stresses induced by bending, and it is therefore easily seen that no formula is rational which would give a stress intensity, on the most strained fiber of the material, exceeding the elastic limit. For example, we may load any column up to a point where the fiber stress "at the most distant point from the neutral axis" reaches the elastic limit, and obtain a certain definite deflection with the load. Any load, however minute, past this point will cause the beam to give, and, as seen in Fig. 1, this

increases the bending arm, which, in turn, increases the fiber stress still more and collapse occurs.

Considering, then, the connecting rod as a long column with pin ends, we have a condition as shown in Fig. 2 A.



Euler's theory, however, is deduced from a column with one end fixed as shown in Fig. 1, and the resulting formulation applied to a column with pin ends, by considering the point of maximum deflection as the origin, and one end fixed at this point. For this condition the length under consideration is one-half the total length of the rod.

In Fig. 1, let AE represent a column fixed at one end, A, and with a pin end at E. Under the action of an axial load this column will take up a new position, AF. Considering A as the origin, we see that at any point, as CD, we have, under the action of the load P, a bending moment equal to

$$P(z-y) = M.$$

From the relationship of slope and deflection we know

$$(1) \quad \frac{M}{EI} = \frac{P(z-y)}{EI} = \frac{d^2y}{dx^2},$$

where I is the moment of inertia of the cross section, which is here assumed to be uniform; E is Young's modulus, and equal to 30,000,000 for iron and steel.

Rearranging (1) we get

$$\frac{d^2y}{dx^2} + \frac{P}{EI}y = \frac{P}{EI}z. \qquad (2)$$

Solving this differential equation, we obtain

$$y = z + B\cos\sqrt{\frac{P}{EI}}x + C\sin\sqrt{\frac{P}{EI}}x, \qquad (3)$$

in which B and C are integration constants and readily determined.

In Fig. 1 we see that when $x = o$, $y = o$, and equation 3 becomes

$$o = z + B,$$
$$\text{or } B = -z.$$

We also see that when $x = o$ there is no deflection, and hence no slope, and

$$\frac{dy}{dx} = o.$$

By differentiating equation (3)

$$\frac{dy}{dx} = \sqrt{\frac{P}{EI}}\left(-B\sin\sqrt{\frac{P}{EI}}x + C\cos\sqrt{\frac{P}{EI}}x\right),$$

and substituting $\frac{dy}{dx} = o$, we get

$$o = \sqrt{\frac{P}{EI}}(o - C) \text{ or } C = o, \qquad (4)$$

$$y = z\left(1 - \cos x\sqrt{\frac{P}{EI}}\right),$$

In Fig. 1 we see that when $x = l$, $y = z$ and, from equation (4)

$$z = z - z\cos l\sqrt{\frac{P}{EI}},$$

$z \cos l\sqrt{\frac{P}{EI}}$ must therefore be zero, and $l\sqrt{\frac{P}{EI}}$ must equal $\frac{n\pi}{2}$; n being any odd number.

The problem now is to find the least load necessary to bend the column a distance z, that is to say the value of $\frac{n\pi}{2}$, which will make P a minimum, evidently $\frac{\pi}{2}$,

$$\text{and } l\sqrt{\frac{P}{EI}} = \frac{\pi}{2},$$

$$\text{(5)} \quad \ldots\ldots\ldots, \text{ or } P = \frac{\pi^2 EI}{4l^2}.$$

Now $I = k^2A$, where k^2 is the squared radius of gyration of the section, and A the area.

Substituting we get

$$\text{(6)} \quad \ldots\ldots \quad \frac{P}{A} = \frac{\pi^2 E}{4}\left(\frac{k}{l}\right)^2.$$

In the case of a connecting rod we have the case of a column with two pin ends, as represented in Fig. 2A, and in this case, taking the origin at o we will consider each half as a column fixed at one end as above, and the length being $\frac{l}{2}$ we get

$$\text{(7)} \quad \ldots\ldots \quad \frac{P}{A} = \pi^2 E\left(\frac{k}{l}\right)^2.$$

If both ends were fixed, the length becomes $\frac{l}{4}$, and

$$\text{(8)} \quad \ldots\ldots \quad \frac{P}{A} = 4\pi^2 E\left(\frac{k}{l}\right)^2 \text{ (see Fig. 2 B);}$$

or in general

$$\text{(9)} \quad \ldots\ldots \quad \frac{P}{A} = m\pi^2 E\left(\frac{k}{l}\right)^2,$$

where m is a constant depending upon end conditions and equal to 1 for the connecting rod.

Let us now examine Euler's formula by means of a concrete

example. We will consider a steel column of fifteen-inch length and one-inch diameter. The steel has a tensile strength of 90,000 pounds and an elastic limit of 47,000 pounds per square inch. Rod has pin ends.

$$P = mEI\frac{\pi^2}{l^2},$$

$$P = \frac{1 \times 30 \times 10^6 \times .0491\, d^4 \times \pi^2}{15 \times 15} = 66{,}000 \text{ pounds.}$$

and

$$\frac{P}{A} = \frac{66{,}000}{.7854} = 84{,}000 \text{ pounds per square inch,}$$

or nearly twice the elastic limit.

As mentioned before, this would show that a column could be strained past its elastic limit without failure, which is manifestly untrue, and this formula will not hold for columns of this length. We now will find the limiting length for which this column formula might possibly be used. We have stated that at no place may the fiber stress exceed the elastic limit, so, assuming $\frac{P}{A} = 47{,}000$ pounds per square inch, we find from

$$\frac{P}{A} = \pi^2 E \left(\frac{k}{l}\right)^2,$$

$$47{,}000 = 9.87 \times 30 \times 10^6 \times \left(\frac{k}{l}\right)^2,$$

$$\frac{l}{k} = 79.5.$$

Thus we see that this formula would hold only when the ratio of $\frac{l}{k}$ exceeds 80, or about 30 diameters for round sections. A plot of this formula, with $\frac{P}{A}$ and $\frac{l}{k}$ as ordinates, is given in Fig. 3. Here again we see that the stress passes the elastic limit at about 80, and increases rapidly as the ratio of $\frac{l}{k}$ decreases. This would seem to show that short rods would bear an

infinite load without failure, which is not true. This can be explained by the fact that in short columns the failure by direct compression becomes the controlling factor.

RANKINE'S AND RITTER'S THEORY.

Rankine, by using a combination of Euler's formulation for long columns and the formula for failure by direct crushing, obtained a formula which is extremely satisfactory in that the curve passes through the elastic limit and agrees well with experimental data for rods of all lengths.

Calling the ultimate load as obtained by Euler's formula, P_e, and the ultimate load for an extremely short column in compression, P_c, the crippling load, P, for any column, is given by

$$(1) \quad \frac{1}{P} = \frac{1}{P_e} + \frac{1}{P_c}.$$

This certainly appears rational, for as the ratio of $\frac{l}{k}$ decreases P_e increases rapidly, and $\frac{1}{P_e}$ becomes negligible, and $P = P_c$.

Remembering that $P_c = f_c A$, we get from equation (1),

$$(2) \quad P = \frac{1}{\frac{1}{f_c A} + \frac{l^2}{\pi^2 EI}} = \frac{f_c A}{1 + \frac{f_c l^2}{\pi^2 E k^2}}.$$

$$(3) \quad P = \frac{f_c A}{1 + a\left(\frac{l}{k}\right)^2};$$

where

$$a = \frac{f_c}{\pi^2 E}.$$

Rankine uses values of "a" determined experimentally while Ritter uses the computed value. The usual value of a for steel is usually taken to be $\frac{1}{7,500}$. A plot of this equation is also given in Fig. 3.

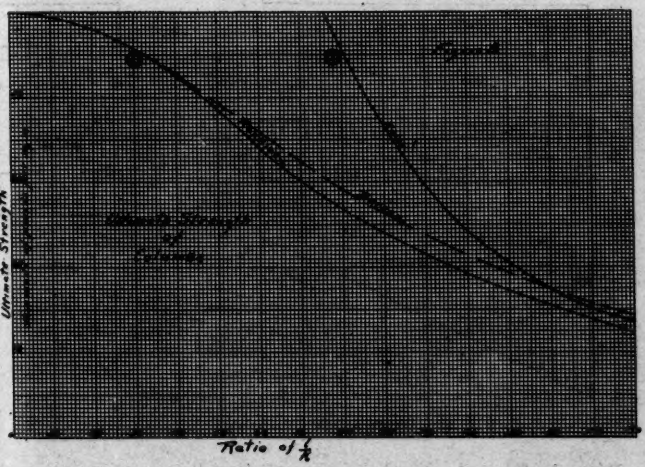
There are several other formulae in use giving more or less

satisfactory results. Johnson's Parabolic Formula is probably the best of these, and is

$$\frac{P}{A} = f_c - b\left(\frac{l}{k}\right)^2,$$

where b is $\frac{f_c^2}{64E}$ for pin ends or $\frac{f_c^2}{100E}$ for fixed ends.

This formula is, however, purely empirical.



From what has been said it is seen that while Rankine's formula is good only when the experimental value of "a" has been determined, Ritter's formula is reliable for all materials as soon as the physical tests of the material to be used have been run. For example, the physical test gives us at once the value of f, the elastic limit, and e, the elongation, and

$$E = \frac{\frac{P}{A}}{e}.$$

The stress diagrams according to the various formulae are shown in Fig. 4.

Euler considers bending alone, which gives a stress distribution through the section as shown in Fig. 4A.

Ritter and Rankine, in addition to this stress, consider also a direct compression which would give a diagram similar to 4B. The additive effect is shown in 4C, which is probably close to the true stress distribution and $f_c''' = f_c + f_c''$.

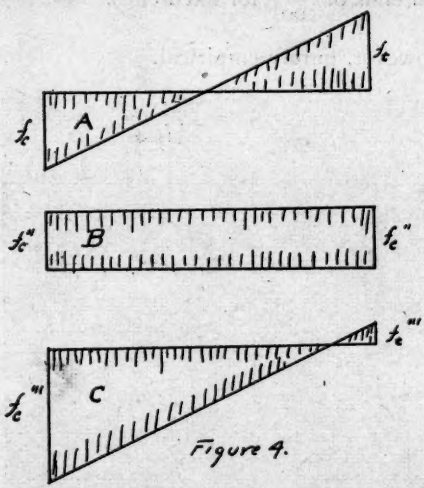


Figure 4.

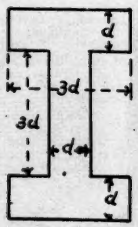


Figure 5.

RITTER'S FORMULA AS APPLIED TO DESIGN.

Ritter's formula in the form given above is not directly applicable to design, but becomes so by a very simple operation:

$$P' = \frac{fA}{1 + \frac{f}{m\pi^2 E}\left(\frac{l}{k}\right)^2}.$$

$$\text{Let } \frac{P'}{\text{constant}} = \text{working load} = \frac{\frac{f}{k}A}{1 + \frac{f}{m\pi^2 E}\left(\frac{l}{k}\right)^2}.$$

This constant is usually between 5 and 6 in practice. While the equation given above gives extremely satisfactory results in practice, it is not easily used in this shape for design,

due to the fact that A and k are both unknown. However, if we can decide upon the shape of the cross section only, the formula is directly applicable.

For example, if we select a round section, solid rod, then

$$A = \frac{\pi d^2}{4} = ad^2 \text{ and } k^2 = \frac{d^2}{16} = \frac{d^2}{b}.$$

If now we let $c = \frac{f}{m\pi^2 E}$, $a = \frac{\pi}{4}$, and $b = 16$, and substitute in this formula, we get

$$(1) \quad \ldots \ldots \quad P = \frac{\left(\frac{f}{k}\right) ad^2}{1 + cb\frac{l^2}{d^2}}, \text{ or}$$

$$(2) \quad \ldots \quad d = \sqrt{\left(\frac{P}{\frac{2fa}{\text{const}}}\right)\left(\frac{1 + \sqrt{1 + \frac{2bcl^2}{P}}}{\frac{2fa}{\text{const}}}\right)},$$

and this becomes for this rod of steel, $f = 47{,}000$,

$$d = \sqrt{\frac{P}{\frac{f + 1.57}{\text{const}}}\left(\frac{1 + \sqrt{1 + \frac{l^2}{\frac{P}{200}}}}{\frac{1.57\ f}{\text{const}}}\right)},$$

The constants a, b and c for any other section are determined in a similar manner; for example, if we desire an I-beam construction similar to Fig. 5, after deciding on the proportion we get

$$A = 9d^2 \text{ or } ad^2;$$
$$a = 9;$$
$$I = \frac{A\ h^2}{6.66}, \text{ but } h = 5d;$$
$$I = \frac{25\ ad^4}{6.66} = \frac{25 \times 9 \times d^4}{6.66} = bd^4;$$
$$b = \frac{225}{6.66}.$$

Let us now compare the following formula, which is largely used,

$$P = \frac{f_t A}{1 + .000111 \frac{l^2}{k^2}},$$

to Ritter's

$$P = \frac{fA}{1 + \frac{f}{m\pi^2 E} \frac{l^2}{k^2}}.$$

The first thing to be noticed is that the "f" used in the first formula is the allowable stress and empirical, while in Ritter's the stress is that at the elastic limit. A plot of the first formula is given in Fig. 3, and while it is not known by what method this formula is deduced, the close agreement as to shape leads us to suspect that it is adopted from Rankine's formula, with some slight change in the value of "a", and therefore empirical. We can see that while this formula is satisfactory for steel it would not be as satisfactory with other metals as the more logical deduction of Ritter. From the curves it is also seen that the stress as determined by Ritter's formula is greater than that obtained by the first formula for ratios of $\frac{l}{k}$ greater than 30.

To show this difference we will make a direct comparison in a rod approximating those of a battleship.

From the specifications for steel we find that the allowable stress is 8,000 pounds per square inch, and that the steel used for this purpose has a minimum tensile strength of 50,000 pounds per square inch.

Taking a hollow rod, 8 inches outside and 4 inches inside diameter, 96 inches long, we find, using Ritter's formula,

$$P = \frac{\frac{f}{\text{const}} A}{1 + \frac{f}{m\pi^2 E}\left(\frac{l}{k}\right)^2};$$

$$\frac{f}{\text{const}} = \frac{P}{A}\left(1 + \frac{f}{m\pi^2 E}\left(\frac{l}{k}\right)^2\right);$$

$$= \frac{280{,}000}{37}\left(1 + \frac{50{,}000 \times 96^2}{1 \times \pi^2 \times 30 \times 10^6 \times 5}\right);$$

$$= 9{,}950 \text{ pounds per square inch};$$

$$\text{constant} = \frac{50{,}000}{9{,}950} = 5.$$

The first formula gives in the same case

$$\frac{R}{A} = \frac{f_c}{1 + .000111\frac{l^2}{k^2}};$$

$$f = \frac{280{,}000}{37}\left(1 + .000111\frac{96^2}{5}\right) = 9{,}400 \text{ lbs. per square inch.}$$

This is a quite remarkable agreement, but, of course, is not as close if we use any but this particular steel.

STRESSES DUE TO CENTRIFUGAL FORCE.

We have so far considered only those stresses in the rod due to a compressive load, and the question next to decide is, Have we taken account of all stresses? A short consideration of the conditions will show us that we have not. There is another force which becomes more and more important as we increase the engine speed, and amounts normally to at least ten per cent. of the direct compressive stress, and is additive to this stress.

If we consider the connecting rod in motion, we see at once that one end has the characteristics of a rotating part, while

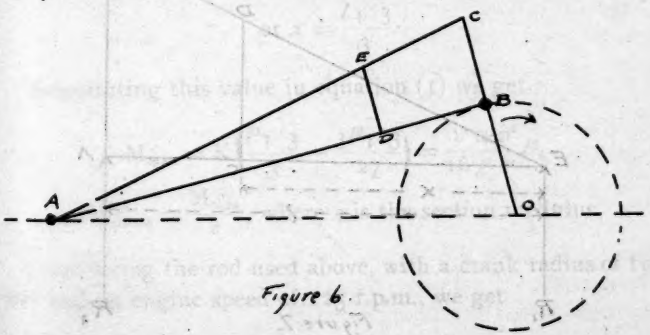


Figure 6.

the end has those of a reciprocating part. All intermediate sections have the characteristics of both.

In Fig. 6, consider the acceleration in the parts when the crank is at right angles to the rod. We see that the acceleration of A is practically zero. The acceleration due to centrifugal force is in the direction BC. If, then, we lay off BC to scale equal to this acceleration and join A and C, we have a curve showing the acceleration at each point of the rod.

The induced force is equal, of course, to the mass times the acceleration, *i.e.*,

$$F = m\alpha.$$

Now if w = weight of rod, in pounds per inch length;

l = length, in inches;

r = radius of crank, in feet;

v = velocity of crank, in feet per second;

W = total load on rod, that is, the area of the loading diagram ABc, we find the ordinate Bc,

$$Bc = \frac{w}{g}\frac{v^2}{r},$$

and

$$W = \frac{w}{g}\frac{v^2}{r}\frac{l}{2}.$$

We can consider the connecting rod as a beam supported at the ends and subjected to a triangular load, Fig. 7. Since the

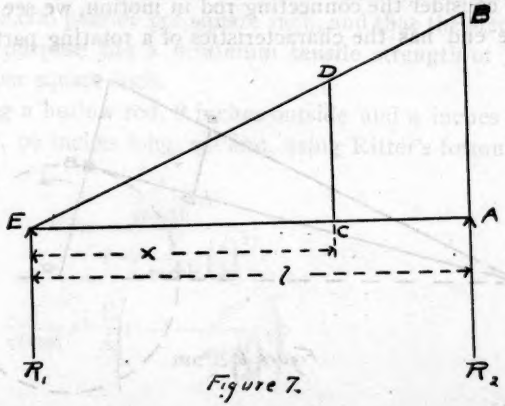


Figure 7.

center of gravity of this load is located at a point one-third the distance from A to E, the reactions on the supports must be

$$R_1 = \frac{w}{3},\ R_2 = \frac{2w}{3}.$$

Take any point C, a distance x from E, and by proportion the ordinate DC equals

$$\frac{w}{g}\frac{v^2}{r}\frac{x}{l}.$$

The load on section EC is DC $\times$ $\frac{EC}{2}$, or $\frac{w}{g}\frac{v^2}{r}\frac{x}{l}\frac{x}{2}$, and the center of gravity is at a distance $\frac{x}{3}$ from C.

Taking moments about C we find that the bending moment

$$M = \frac{W}{3}x - \frac{w}{g}\frac{v^2}{r}\frac{x^2}{2l}\frac{x}{3}, \text{ or}$$

$$M = \frac{1}{6}\frac{w}{g}\frac{v^2}{r}\left(lx - \frac{x^3}{l}\right), \text{ or}$$

(1) . . . . . $M = K(l^2x - x^3)$.

The question now is where is this bending moment a maximum?

Differentiating (1) with respect to x we get

(2) . $dM = K(l^2 - 3x^2)\,dx$, equating to zero $l^2 - 3x^2 = 0$,

$$\text{or } x = \frac{l\sqrt{3}}{3}.$$

Substituting this value in equation (1) we get

(3) . . $M_{max} = K\left(\frac{l^3\sqrt{3}}{3} - \frac{3l^3\sqrt{3}}{27}\right) = \frac{1}{16}\frac{w}{g}\frac{v^2}{r}l^2$,

and $f_{max} = \frac{M_{max}}{z}$, where z is the section modulus.

Considering the rod used above, with a crank radius of two feet and an engine speed of 125 r.p.m., we get

$$w = (50.26 = 12.57) .254 = 9.6 \text{ pounds};$$

$$v = \frac{125}{60} \times 12.57 = 26.2 \text{ feet per second};$$

$$z = \frac{AD^2 - ad^2}{8D} = 47.2.$$

From equation (3)

$$M = \frac{1}{16} \times \frac{9.6}{32} \times \frac{\overline{26.2^2}}{2} \times \overline{96^2} = 59{,}200 \text{ pounds.}$$

$$f = \frac{M}{z} = \frac{59{,}200}{47.2} = 1{,}250 \text{ pounds per square inch.}$$

Ritter's formula gave the stress due to column action of 9,950 pounds per square inch, and from the above we have an additional stress of 1,250 pounds per square inch due to centrifugal action, or a total of 11,200 pounds, thirteen per cent. greater than that given by either the Bureau or Ritter's formulas.

As the stress due to centrifugal force varies as the square of the velocity, this force becomes important in high-speed marine engines.

From what has been said it is apparent that in investigating the stresses in any connecting rod, especially in cases of failure, we should base such an analysis upon the physical test of the original forging, using Ritter's formula, and correcting this result for the operating conditions at the time of fracture, or upon the maximum engine speed. The original data should be used in preference to data from a test piece cut from a broken or defective rod, due to the fact that a metal strained slightly beyond the elastic limit will show a higher elastic limit on a subsequent tensile test.

Finally, it must be remembered that in examining a broken rod, the cause of failure should not be judged from the "appearance of fracture." Recently I have seen examples of broken parts where the break was attributed to crystallization. *The crystalline structure of a metal does not change under stress below the elastic limit. Fatigue is possible, but crystallization never.* Examine for stress first. The metal has probably been tested and found satisfactory.

NOTES.

THE FUTURE OF SHIP PROPULSION.

Since the days of the Romans, whose motto was that the three essentials to civilization were roads, more roads, and "still some," the close connection between cheap transport and the amenities and, indeed, the necessities, of society has been realized, but much more clearly by the engineer and business man than by the public at large. In fact, ignorant publicists have been known to aver that the masses of the people have derived no benefit whatever from the work of the engineer, which has made it possible to transport meat some 8,000 to 12,000 miles, from New Zealand or the Argentine, and enables wheat grown west of Winnipeg to find its way into British silos. One hundred and fifty years ago 20 miles was not far from the limit beyond which the cheaper kinds of food could not be profitably conveyed by land, and though sea freights were much cheaper than land carriage, imported goods could not find their way far from their port of entry, save at rates prohibitive to all but the more costly class of commodities. The contrast with the present day is extraordinary, and though much, no doubt, yet remains to be done, it is most unlikely that any equally striking step forward can be made in the years to come. All we can look for is a steady step-by-step advance, and this applies with special emphasis to the problem of marine transport. In normal times this is extraordinarily cheap, but there is nevertheless a sustained endeavor to reduce still further operating costs. A few years ago there was a feeling somewhat prevalent that for marine propulsion the future lay with the Diesel engine, but further experience has proved that this type of motor is desirable only in somewhat special circumstances. Indeed, some engineers claim that even where oil fuel is available, it is now more profitable to use this under boilers than to instal Diesel engines even of moderate size. Of course, in the case of very large powers, the Diesel engine is admittedly inferior, all things considered, to the steam turbine. It is, nevertheless, undoubtedly of interest to note that Messrs. Burmeister and Wain are reported to have received orders for motor ships which will keep them occupied till the end of 1921. Included in the list are six ships, of 12,800 tons dead weight, which are to have motors of 5,300 horsepower.

The very high economies realized with the steam turbine in land installations have naturally initiated inquiries into how far similar results may be obtainable at sea. A few years ago engineers had by no means abandoned the hope of devising an efficient high-speed propeller, thus abolishing the clog which the standard type had established on the speed of rotation needed to secure an efficient turbine. This anticipation appears now to have been abandoned, and, indeed, the recent elucidation of the propeller theory, which has been effected by those interested in aerial navigation, gives reason for belief that the idea is little likely to be resuscitated. All hopes are now centered upon the introduction of some kind of reduction gear, mechanical, electrical or hydraulic, between the turbine and the propeller shaft. The first named has the advantage of a very high efficiency. The electric and the hydraulic systems are less

efficient *per se*, but have some compensating advantages. Moreover, the claim is made that they will permit of the use of a more economical steam plant than can be adopted with mechanical gears.

A most instructive comparison of the relative advantages of different methods of propelling a 21,000-ton 19¼-knot liner is given in a paper recently read before the Institution of Engineers and Shipbuilders in Scotland by Mr. James Dornan, and a consideration of which we have deferred until it had been discussed fully at the Institution. Papers of this kind are exceedingly valuable, and necessitate a very large amount of labor, of which the outcome is often merely the addition of a few rows of figures in the tabulated results. The compact manner in which the outcome of toilsome calculations can thus be expressed fails to draw due attention to the great care and thought that have been expended in the compilation.

The worth of such a paper is by no means dependent upon the absolute precision of the assumptions necessarily made in the course of the work. It was a politician, concerned, like his kind, for his consistency, who formulated the view that you should never prophesy unless you know. If this motto were adopted by engineers the world would be at a standstill, since all improvements necessarily involve some measure of vaticination. Certain assumptions have to be made, as to the sufficiency of which only experience can decide. In any paper forecasting the future certain risks must accordingly be taken, and as no one engineer has a universal experience, his particular assumptions may not prove acceptable to every individual of his audience, each of which has his own special experience. Indeed, this circumstance adds to the value of papers such as we are considering, as in the discussion each specialist is able to support his own claims.

The two main assumptions made by Mr. Dornan were that tooth pressures and pitch-line speeds have already reached their limits in large marine gears, and that highly superheated steam cannot be used in installations in which reversing turbines are employed, and it was on these points that the criticism of his paper largely centered. On the above hypotheses Mr. Dornan has worked out comparative results for seven different types of propelling machinery suitable for a liner measuring 600 feet by 72 feet by 27 feet draught, displacing 21,000 tons, and having a service speed of 19¼ knots. His results are summarized in the table.

A noteworthy point is the excellent showing made by the reciprocators using superheated steam. These engines are, it will be seen, credited with a better performance than the corresponding turbines, which is attributable to the higher superheat used. In this connection it may be observed that had high superheats come into general use for reciprocating engines twenty-five to thirty years ago, the task successfully carried through by Sir Charles Parsons might well have proved impossible. His turbine is essentially a big-unit machine, and his ability to develop it in the days of small units was largely due to the relatively high steam consumption of the reciprocating engines then available. This defect has, in the outcome, been to the advantage of the world, as the experimental work necessary to the successful development of the Parsons turbine could never have been financed if it had had to be carried out on large units.

The high position occupied by the hydraulic gear in Mr. Dornan's table is due, in the first place, to the ratio of reduction being suitable for this system, which is well known to be inapplicable for speed ratios greater than some 6 to 1, and, secondly, to the high superheat used. With a propeller speed of 200 revolutions per minute, it is, moreover, credited with the same propeller efficiency as the geared-turbine machinery, in which the speed is 160 revolutions per minute. Had this figure been increased to that adopted for the hydraulic gear, a smaller

COMPARATIVE DATA OF DIFFERENT TYPES OF PROPELLING MACHINERY FOR A STEAMSHIP 600 FEET LONG, 72 FEET BEAM, 27 FEET DRAUGHT, 21,000 TONS DISPLACEMENT, AND 19½ KNOTS SPEED.

Design.....	A.	B.	C.	D.	E.	F.	G.
.....	Two-shaft quadruple engines.	Four-shaft direct turbines.	Two-shaft quadruple engines. (superheat).	Four-shaft direct turbines (superheat).	Two-shaft hydraulic gears.	Two-shaft mechanical gears.	Two-shaft turbo-electric gears (Ljungström).
Number of shafts	2	4	2	4	2	2	2
Revolutions per min. (propeller shaft)	85	290	85	290	200	100	85
Indicated horsepower	21,650	.....	21,650	.....	.....	.....	.....
Shaft horsepower	.....	21,800	.....	21,800	21,300	21,350	20,000
Revolutions per minute (turbines)	.....	290	.....	290	1,000	1,800	3,000
Working pressure, lbs. per square inch	210	200	210	200	200	200	200
Superheat, degrees F	Nil.	Nil.	200	100	200	100	260
Vacuum, inches	27	28	27	28	28½	28½	28½
Steam per hour—main engines, pounds	274,400	258,400	225,000	238,000	211,050	220,200	167,240
—auxiliaries, pounds	43,600	43,600	36,000	38,000	35,000	35,000	29,000
Coal per horsepower hour, pounds	1.47	1.385	1.257	1.292	1.205	1.218	1.032
B.t.u. per horsepower hour	20,600	19,400	17,600	18,100	16,900	17,050	14,500
Economy over Design A, per cent	.....	5	14½	11½	19	18	35
Boilers, double-ended	6	6	6	6	6	6	4
single-ended	4	4	3	3	2	2	3
Heating surface, square feet	52,000	52,000	48,750	48,750	45,500	45,520	35,750
Grate area, square feet	1,232	1,232	1,155	1,155	1,078	1,078	847
Weight of machinery, tons	3,675	2,965	3,610	2,900	2,445	2,605	2,310
plus coal, tons	6,403	5,557	5,946	5,316	4,653	4,837	4,086
Saving in machinery space, feet	.....	15	18	21	60	51	72
Cubic capacity gained, cubic feet	.....	18,100	21,200	24,250	64,900	56,970	80,600
Coal used per annum, tons	51,150	48,600	43,800	45,300	41,400	41,850	33,300
Extra deadweight carried yearly, tons	.....	20,304	10,968	26,088	42,000	37,584	55,608
Value of coal saved, pounds	.....	2,550	7,350	5,850	9,750	9,300	17,850
Value of extra deadweight, pounds	.....	14,213	7,678	18,261	29,400	26,308	38,925
Total saving, pounds	.....	16,763	15,028	24,111	39,150	35,608	56,775
Cost of ship, pounds	650,000	650,000	668,500	670,500	673,000	672,000	678,000

In all the comparisons made A is taken as the standard of reference.

or more efficient turbine could have been used. In Mr. Dornan's view, however, an alteration of this kind would have involved dangerous tooth pressures and pitch-line speeds.

It is on this point that criticism naturally fixed. Experience with high-speed gears is still somewhat limited, and many more breakages must occur before engineers can be certain that the limit of safety has been reached. With the relatively large pitches used in some of the earlier plants, a breakage occurred in one case with a tooth pressure equal to about 1,100 pounds per inch width of face. The pitch in this case was about 4/5 inch, and it is understood that the teeth were of unsuitable form. According to Mr. J. H. Gibson, the standard pitch for large marine gears is now 0.583 inch, and the wheel teeth have very short addenda. It is understood that in certain ships now building the gear-driven propeller shafts are to transmit nearly twice the power per pinion specified by Mr. Dornan. It is to be hoped that full data as to these remarkable installations will be available at a later date, so that comparison may be made with the figures brought forward by Mr. J. H. Macalpine in his most interesting contribution to the discussion on Mr. Dornan's paper. Mr. Macalpine claims many advantages for the method of mounting gears in a floating frame; but these claims have not, so far, been admitted by British engineers, who, in general, are of opinion that, with accurately-cut teeth of suitable shape, frames of the type in question are superfluous, and the strength of their conviction is well evidenced by the remarkable sets of gears just referred to The idea embodied in the floating frame is that the pinion shall be located by the teeth of the gear rather than by fixed bearings At first sight, however, any self-adjustment of the frame to compensate for errors in the teeth would appear to be impossible, owing to the very high speed of the pitch line and the considerable inertia of the pinion and its frame. If the error in question were confined to a single tooth, it would seem that this conclusion would be indefeasible; but the experiments of Sir Charles Parsons do undoubtedly put rather a different aspect on the matter. He found by actual measurement that the errors in well-cut gears were periodic in character. This observation much improves the case for the floating frame, since such periodic errors would presumably follow approximately a sine law. The frame therefore would not have to adjust itself violently and instantaneously, as it would with the error confined to a single tooth or a single region of the wheel, but would gently oscillate from a position in which the positive errors in the teeth were a maximum to that in which they were a minimum. The change of motion being made at these maximum and minimum values would greatly minimize the effect of the inertia of the pinion.

As is well known, Sir Charles Parsons solved the problem by abolishing the periodic errors. From Mr. Macalpine's remarks, it would seem that the floating frame has proved an equally effective solution, and the figures he gives are of the greatest interest, and indicate that the weights and dimensions assumed by Mr. Dornan as necessary when mechanical gearing is introduced may be very materially reduced. Mr. Macalpine states that the Westinghouse Machine Company are making gears for two foreign warships, each to be fitted with machinery developing 22,000 horsepower. The guarantees with dry saturated steam at 225 pounds per square inch are as follow:—

	Consumption per shaft horsepower.
Full power, pounds	10.8
80 to 85 per cent., pounds	10.5
25 per cent., pounds	11.5
15 per cent., pounds	12.8

At 15 per cent. of full power, corresponding to about half speed, the consumption per shaft horsepower is only 18.5 per cent. greater than at full power. He states that for the case taken in Mr. Dornan's paper, where 5,100 horsepower are to be transmitted per pinion, the following proportions would suffice:—Pinion diameter, 8 inches; wheel diameter, 148 inches; revolutions per minute of pinion, 2,500; revolutions per minute of propeller shaft, 135.1. The length of the pinion would be 4 feet 8 inches, as compared with the 7 feet 6 inches or 8 feet assumed by Mr. Dornan. Mr. Macalpine further observes that the modifications he suggests in Mr. Dornan's figures would enable the size of the turbines to be reduced, so that the mean diameter at the low-pressure end would be only 4 feet 7 inches. We are inclined to believe, however, that this change would give a steam way somewhat more restricted than is desirable, unless it be intended to use double-flow turbines. For a 28½-inch vacuum, the mean diameter in inches of the last row of blades should not be less than about $0.63 \sqrt{\text{S.H.P.}}$, or, say, 64 inches. Of course, there is no absolutely fixed rules in this matter, since different engineers have different views as to the loss permissible by "carry over" to the exhaust.

By taking advantage of the improvements now realized in mechanical reduction gears, Mr. Macalpine concludes that the advantages claimed in Mr. Dornan's paper for the turbo-electric drive are transferred to its rival, which would occupy a length 16 feet 4 inches less than the turbo-electric drive, in place of the 21 feet more with which it is credited in the paper. The turbo-electric installation described by Mr. Dornan has a special interest of its own. It is proposed to use Ljungström turbines running at 3,000 revolutions per minute, and supplying current to electric motors which drive the propeller shafts through mechanical reduction gears. One reduction of speed is effected electrically, and the remainder by the mechanical gears, the advantage of the system being the large reduction ratio practicable without the reduction in power factor which would be involved in the use of multipole motors.

A Ljungström turbine capable of developing 5,000 kw. at over-load has already been constructed, and undergone its preliminary tests. The highest load reached in these was about 2,000 kw., the steam consumption at this load being 11 pounds per kw.-hour, with a steam supply at 15 atmospheres absolute, and a superheat of about 280 degrees F. The vacuum was extremely high, so that there would probably be no very great improvement in the water rate when the output was doubled. According to the curves given in the "Teknisk Tidskrift" the anticipated consumption at 5,000 kw. is about 10.8 pounds per kw.-hour. In his paper Mr. Dornan has assumed that with turbines at 10,000 horsepower a water rate of 7.5 pounds per shaft-horsepower hour will be recorded, and has presumed that it will be possible to run the turbine at 3,000 revolutions per minute. The limit to the output of a Ljungström turbine at a given speed appears to depend on the critical velocity of the largest ring of blades. The ouptput depends upon the length of this row, which must not be so great that its natural periodicity coincides with the rate of revolution of the turbine. The linear velocity feasible for this row of blades is, of course, fixed by the permissible centrifugal stresses. According to Mr. Konrad Anderson stresses of 16 tons per square inch are admitted in De Laval wheels, and a rough estimate of the size of the 5,000-kw. unit, illustrated in the "Teknisk Tidskrift," conveys the impression that the centrifugal stresses in this machine are of somewhat the same order.

The advantages of the Ljungström system are most marked in the case of units of what are now considered moderate outputs. We gather from Mr. Dornan's reply to the discussion that the 5,000-kw. set above referred to has a "coefficient" of 300,000, which is remarkably high for a turbine of 5,000 kw. only. Even the great Chicago turbine has a coefficient of

but 216,000. High coefficients can no doubt be obtained with axial-flow machines of moderate output at the price of introducing velocity-compounded impulse stages at the high-pressure end, but it is not possible to attain in this way the efficiencies realizable with the "straight" reaction system. On this head Mr. Macalpine's criticisms of Mr. Dornan's paper seem beside the point, as it will never be possible to reach with disc and drum machines the same limit of efficiency attainable with the straight reaction type. When constructed on the axial-flow principle, however, any close approach to the theoretical limit is commercially impracticable in the case of small machines, but this difficulty is surmounted in the Ljungström turbine, with which remarkable results have been secured in the S. S. *Mjölner* with a plant developing but 843.4 shaft horsepower.

It will be noted that much of the advantage claimed for Mr. Dornan's electrically-geared drive is to be attributable to the high superheat of 260 degrees, whilst 100 degrees superheat is all that he allows to the geared turbine. No doubt there have been difficulties met with in using superheat at sea with existing patterns of marine turbine. In fact, at the last meeting of the Institute of Metals, Admiral Oram observed that they had adopted superheat but sparingly in the Navy, and that he personally should prefer to be without it entirely. This view, which is founded on practical experience, is the more noteworthy in that the superheat has in all cases been very moderate, and the turbines to which it was applied of a type generally presumed to be free from troubles due to differential expansion and the like. On the *Mjölner* a superheat of 190 degrees F. has, we understand, been used and has given no trouble during the many months the vessel has been in service. Of course, the Ljungström type is particularly well adapted to the use of high-temperature steam. It is almost perfectly symmetrical, so that it must be nearly free from distortion by temperature changes. Moreover, in parts where the temperature is high the steam velocity is low, whilst in disc-and-drum machines high temperatures and high velocities are associated together, which is no doubt largely responsible for such trouble as has been found in securing a satisfactory metal for the blading. It would, however, be rash to assume that these difficulties are insuperable, particularly if it is merely on account of the reversing turbine that Mr. Dornan has limited the superheat to 100 degrees F. in proportioning his geared turbine. In this connection it is of interest to note that the builders of the Ljungström turbine are themselves experimenting with all-gear drives, thus suppressing the losses necessarily associated with the electrical plant Some of this loss might, no doubt, be recovered by a greater use of current to drive the auxiliaries now commonly actuated by "steam-eaters," and in view of the very large proportion which the auxiliary steam frequently bears to that entering the main propelling machinery, some satisfactory method of economizing auxiliary steam is badly desiderated.

As to the relative advantages of mechanical and electric drives, some figures contributed by Mr. Macalpine should be put on record here. He stated that very careful comparative estimates of the weights involved for a 10,000-horsepower twin-screw ship had been got out by the Westinghouse Company.

Type of machinery.	Machinery weights (exclusive of boilers and auxiliaries).
Geared turbines, pounds	274,000
Turbo-electric drive (geared generators), pounds	304,000
Turbo-electric drive (direct-connected generator), pounds	423,800

Mr. Macalpine further stated that an estimate made for a gear-driven 28-knot battleship developing 64,000 horsepower showed that the total

weight of turbines, gears, and thrust blocks was but 212 tons, the propeller speed being 420 revolutions per minute. It is proposed to use thrust blocks of the Michell (Kingsbury) type, of which 22 are now at work in America under loads ranging from 350,000 pounds up to 550,000 pounds, in addition to many taking smaller thrusts. In such ships as discussed in Mr. Dornan's paper, the use of these blocks would save, he states, at least 8 feet of length.—"Engineering."

TRIALS OF GEARED TURBINE PROPELLING MACHINERY ON FREIGHT STEAMER *PACIFIC*.

VESSEL EQUIPPED WITH CURTIS FIVE-STAGE GEARED TURBINE OF 2,400 HORSEPOWER SUPPLIED WITH SUPERHEATED STEAM FROM SCOTCH BOILERS.

The standardization trials of the steamship *Pacific*, recently built by the Union Iron Works, San Francisco, Cal., for Norwegian owners, were held on November 21, 1915, over the measured-mile trial course off Bluff point, in San Francisco Bay. This vessel is of special interest at this time, as she is one of the first large general cargo steamers of moderate speed built in the United States to be equipped with geared-turbine propelling machinery.

The *Pacific* is 418 feet long over all, 400 feet long between perpendiculars, with a molded beam of 56 feet and a molded depth to upper deck of 32 feet. In light condition, with a draught of 8 feet 3½ inches, the displacement is 3,540 tons, and the tons per inch of immersion 41.17. In loaded condition, with a draught of 25 feet 4 inches, the displacement is 12,500 tons, and the tons per inch of immersion 46.16. The main characteristics of the hull in both the light and loaded conditions are shown in the following table:

	Light	Loaded
Block coefficient	.673	.773
Prismatic coefficient	.712	.790
Area of midship section	435	1,385
Coefficient of midship section	.942	.930
Area of water plane	17,330	19,390
Center of buoyancy above base	4.96	13.8
Transverse metacenter above base	36.4	24.28
Longitudinal metacenter above base	1,890	473

Propulsion of the vessel is by a Curtis 5-stage geared turbine supplied by the General Electric Company, Schenectady, N. Y., and designed to develop 2,400 shaft horsepower at 3,500 revolutions of the turbine and 90 revolutions per minute of the propeller shaft. Steam is supplied at a pressure of 210 pounds per square inch by three single-ended, three-furnace Scotch boilers, 14 feet 9 inches inside diameter and 11 feet long. Each boiler has a heating surface of 2,685 square feet, and is fitted with superheaters of the Union Iron Works Company's type.

The propeller is of the built-up type with a cast-iron hub and four composition blades with a diameter of 16 feet 6 inches, a pitch of 14 feet and an expanded area of 77.04 square feet.

The following tables give the pump data and a list of the auxiliary machinery:

PUMP DATA.

1 Main air pump	Vertical twin beam, 14 ins., 28 ins., 18 ins.
1 Circulating pump	U. I. W. Co.'s centrifugal, 14 ins. dia., 42-in. runner.
1 Main feed pump	De Laval turbine, 37 H. P.
1 Auxiliary feed pump	De Laval turbine, 37 H. P.
1 Fire and bilge pump	Duplex horizontal, 12 ins., 8½ ins., 12 ins.
1 Ballast pump	12 ins., 10½ ins., 12 ins.
1 Trimming pump	10 ins., 7 ins., 10 ins.
1 Sanitary pump	7½ ins., 5 ins., 6 ins.
1 Fresh-water pump	7½ ins., 5 ins., 6 ins.
1 Evaporator pump	4½ ins., 2¾ ins., 4 ins.
1 Engine-room bilge pump	6 ins., 5¾ ins., 6 ins.
2 Fuel-oil pumps	6 ins., 4 ins., 6 ins.

AUXILIARY MACHINERY.

1 Circulating-pump engine	10 ins. × 10 ins.
1 Steam tiller	Brown type, 7½ ins. × 12 ins.
1 Capstan	8 ins. × 10 ins.
1 Windlass	10 ins. × 16 ins.
11 Winches	Ten 7 ins. × 10 ins.; 1-9 ins. × 12 ins.
2 Generators	G. E. Co.'s 15-horsepower turbine, 10 kilowatt.
1 Feed-water heater	U. I. W. Co.'s type.
1 Evaporator	12 ton.
1 Ice machine	Vulcan Iron Works type.

On the standardization trials of the vessel, which were held on November 21, the ship left the builders' wharf at 9:15 A. M., and after the compasses were adjusted, headed for the California city course at 10:05 A. M. The four-hour run was begun at 11 o'clock. Two double runs were made over the course, and then the vessel was headed out for a short run at sea. The four-hour run was finished at 3 P. M., and the vessel landed at the builders' wharf at 4:15. The draught of the ship was 6 feet forward and 16 feet 1 inch aft, making the mean draught 11 feet ½ inch. The fuel oil consumed per shaft horsepower per hour, allowing 10 per cent. for the auxiliaries, figured out at .83 pound. Referring to the indicated horsepower and allowing 10 per cent. for the auxiliaries, the fuel-oil consumption was .73 pound per horsepower-hour. The data taken on these trials are shown in Table I.

TABLE I.—STANDARDIZATION TRIALS, S. S. "PACIFIC."

	Mean of 2 double runs over measured mile.	4-Hour trial at sea.
Speed, knots	13.1	.....
Slip, per cent.	—0.05	.....
Revolutions per minute	89	88.5
Horsepower	2,360	2,355
Boiler pressure, pounds per square inch	207	207
Vacuum, inches	29	29
Temperature, superheated steam, F	420	420
Temperature, saturated steam, F	390	385
Temperature of feed water, F	175	180
Temperature in stack, F	675	672

On December 13 and 14 the *Pacific* made a voyage from Seattle to San Pedro, and during this voyage the data given in Table II were taken.

TABLE II.—DATA TAKEN ON VOYAGE FROM SEATTLE TO SAN PEDRO.

	Dec. 13-14.	Dec. 14.	Dec. 14.	Dec. 14.
Speed, knots	10.88	11.2	10.5	9.8
Revolutions per minute	85.9	86.	81.	77.
Shaft horsepower	2,450	2,450	2,030	1,725
Boiler pressure, pounds per sq. in.	200	200	200	190
Vacuum, inches	29¾	29½	29½	29½
Temperature, superheated steam, F.	410	...	...	...
Temperature, saturated steam, F.	390	...	...	...
Temperature, feed water, F.	208	205	205	210
Temperature in stack, F.	590	630	630	630
Pounds of oil per shaft horsepower hour, allowing 10 per cent. for auxiliaries	0.95	0.97	1.001	1,07

While on the trial trip the *Pacific* was carrying only 500 tons load. Some additional data are available, which were taken on a trip from Seattle after the ship had taken on a full cargo of 8,300 tons, the results of which are given in Table III.

TABLE III.—THREE-HOUR FULL-LOAD TEST AT SEA.

Speed, knots	11.5
Revolutions per minute	87
Draught of vessel, feet	25
Load, tons	8,300
Steam pressure at turbine, pounds per square inch	207
Vacuum, inches	28
Horespower, average	2,655
Pounds of oil per brake horsepower-hour	0.875
British thermal units per brake horsepower-hour	16,420

During the trial run in light condition the reduction gears were noisy, but, with the increased draught on the run in full load condition, the operation of the gears, as regards noise, was greatly improved. The complete immersion of the propeller insured a steady load on the gears, with the result that the noise noticeable in the trial test was gradually reduced, so that it was practically unnoticeable anywhere about the ship, and was a matter of no inconvenience even in the engine room. After the trial trip a larger oil cooler was installed, and it is considered probable that the increased quantity of oil which is now being sent through the gears, due to the use of a larger cooler, has a material effect in giving smoothness in the operation of the gears.

From the data given above it will be noted that a speed of 11.5 knots was obtained at 87 revolutions per minute of the propeller, which indicated that the pitch of the propeller is slightly high for the speed and horsepower of the turbines. This will be remedied in future boats by decreasing the propeller pitch to such a point that a speed of 11.5 knots will be secured at about 90 revolutions. This should result in a gain in the efficiency of the propeller and improve the operation of the ship when lightly loaded.

The propelling machinery of the *Pacific* has proved satisfactory in every respect, and the captain and chief engineer of the vessel have been enthusiastic in their praise of the equipment, and consider that the *Pacific* is by far the most economical vessel of its capacity on the Pacific Coast.

Considering the fact that this design of the turbo-gear propelling machinery is new throughout and that the machinery was installed in less than ten days, and that the ship was put into commission the day after she was finished, and left immediately on a voyage of several months' duration around Cape Horn, it will be appreciated that the turbine and reduction gears have been given a very severe test. The satisfactory results and the economy gained will, therefore, be of much interest to all steamship owners and should pave the way for a wider adoption of this type of machinery in modern slow-speed vessels.—"International Marine Engineering."

NEW LUCKENBACH FREIGHT STEAMERS.

TWO SINGLE-SCREW AND TWO TWIN-SCREW VESSELS TO BE PROPELLED BY CURTIS GEARED TURBINES.

Four large freight steamers are now under construction at the yards of the Fore River Shipbuilding Corporation, Quincy, Mass., for the Luckenbach Steamship Company ,New York. All four of these vessels will be driven by high-speed Curtis turbines, driving the propellers through speed-reduction gears. Two of the vessels are single screw and two twin screw. The single-screw vessels are of 10,000 tons and the twin-screw vesels of 10,500 tons deadweight carrying capacity. They are designed for a speed of 13½ knots on a 28-foot draught. The principal dimensions of the single-screw vessels are as follows:

Length overall, feet	456
Length between perpendiculars, feet and inches	437–6
Beam, molded, feet	57
Depth, molded, feet and inches	41–6

The twin-screw vessels are slightly larger, as shown by the following dimensions:

Length overall, feet	469
Length between perpendiculars, feet	447
Beam, molded, feet and inches	55–11
Depth, molded, feet and inches	40–8

The vessels are built according to designs by Mr. J. L. Luckenbach, vice-president of the Luckenbach Steamship Company, and are classed by Lloyd's and the American Bureau of Shipping.

PROPELLING MACHINERY OF SINGLE-SCREW VESSELS.

The propelling machinery of the single-screw vessels is located amidships and consists of a Curtis turbine, driving a high and low-speed reduction gear designed to deliver 4,000 shaft horsepower at 90 revolutions of the propeller shaft. The maximum shaft horsepower will be 4,500, with the revolutions of the propeller not exceeding 93.

The turbine is designed to operate with a steam pressure of 195 pounds in the steam chest and against a vacuum of 28 inches.

The turbine is so arranged that the ahead and reverse turbines are carried on one shaft and contained in one casing. The astern turbine will be capable of delivering to the shaft not less than 50 per cent. of the power of the ahead turbine. The revolving element will be so designed as to stand an overspeed of 20 per cent. above the full speed.

The total reduction of speed between the turbine and the propeller

shaft will be in two steps, the system employed being what is known as the double-drive type—that is, the pinion will drive two gears, one diametrically opposite the other, thereby eliminating pressure on the pinion bearings and insuring perfect tooth contact throughout the entire guiding surface of the turbine.

The high-speed pinion will be directly connected to the turbine shaft by means of a flexible coupling, so arranged that any actual motion in the gears or pinions will not be transmitted to the turbine shaft. Each wheel will carry a right-and-left helical gear.

The low-speed reduction gear will consist of a large gear wheel driven by two pinions, each pinion being driven through a flexible connection from one of the wheels in the high-speed gear casing.

The propeller is 19 feet diameter with the blades set at a pitch of 16 feet. The pitch, however, is adjustable from 15 feet to 17 feet. The propeller hub is made of cast iron fitted to the propeller shaft with a taper and key and held by a forged-steel nut. The propeller blades are made of manganese bronze.

PROPELLING MACHINERY OF THE TWIN-SCREW VESSELS.

In the twin-screw vessels the propelling machinery, located amidships, consists of two Curtis turbines driving high and low-speed reduction gears, designed to deliver 2,000 shaft horsepower each, at 105 revolutions per minute of the propellers. The total maximum shaft horsepower developed with the revolutions of the propellers not exceeding 109 per minute will be 4,500.

The principal characteristics of the turbines and reduction gears on the twin-screw vessels are the same as those for the single-screw vessels, as explained above. The propellers, however, are solid, 14 feet 6 inches diameter, made of bronze.

BOILERS.

On the single-screw vessels steam is supplied by three Scotch boilers, each 16 feet 6 inches inside diameter, and 12 feet 3 inches long over the heads, with a heating surface of 3,546 square feet. Each boiler has four Morrison corrugated furnaces, 42 inches inside diameter. The boilers are designed for a pressure of 195 pounds per square inch and work under the heated system of forced draft, for which one Sturtevant multivane-blower fan with two engines is located in the engine room. The boilers are arranged to burn fuel oil on the Dahl system, the necessary pumps, filters, heaters, etc., being arranged in the fireroom.

On the twin-screw vessels steam will be supplied by four Heine water-tube boilers of 15,000 square feet heating surface working under forced draft and burning oil on the Dahl system.

AUXILIARIES.

The auxiliaries on both the single and twin-screw vessels are identical. A main condenser of 5,850 square feet of cooling surface is provided, connected to an air pump of the independent wet-and-dry twinplex beam type, the necessary cooling water being supplied by a main centrifugal pump of 18 inches diameter directly connected to a 10-inch by 9-inch single-cylinder vertical engine.

The necessary pumps, which are of the Blake & Knowles type, the high and low feed-water heaters of Griscom-Spencer make, and an auxiliary condenser are located conveniently in the engine room. Abaft of the reduction gear is the thrust bearing, consisting of ten horseshoes with ample wearing surface. The line shafting consists of seven sections with the necessary bearings.

The electric plant consists of two 20-kilowatt General Electric genera-

tor sets, driven by reciprocating engines. There is a 2-ton Brunswick refrigerating machine and a Griscom-Spencer evaporating machine of 35 tons capacity.

For handling cargo there are 24 derricks, served by 24 single-drum winches of the Hyde type. The capstan and windlass on the single-screw vessels will be supplied by the American Engineering Company, while on the twin-screw vessels the capstan and windlass are of the Hyde type. The steering gear is of the Brown tiller type.

The lifeboats are handled under Martin-Freeman davits.—"International Marine Engineering."

PERFORMANCE AND DESIGN OF HIGH-VACUUM SURFACE CONDENSERS.

By Geo. H. Gibson, New York, Member of the Society, and Paul A. Bancel, New York, Junior Member of the Society.

The coefficient of heat transmission in an experimental condenser filled with pure steam appears from recent investigations to be a determinate quantity dependent upon steam temperature, water velocity, diameter of tube and mean water temperature and its value is of the order of 500 to 1,500 B.t.u. per square foot per hour per degree F. difference. In the commercial condenser the high rate of transmission undoubtedly existing for certain parts of the surface is masked by the large amount of relatively idle surface, so that the average coefficient for all the surface figures out much lower—sometimes as low as 50 B.t.u. The size and cost of a condenser to maintain a certain vacuum depend primarily on the extent of the

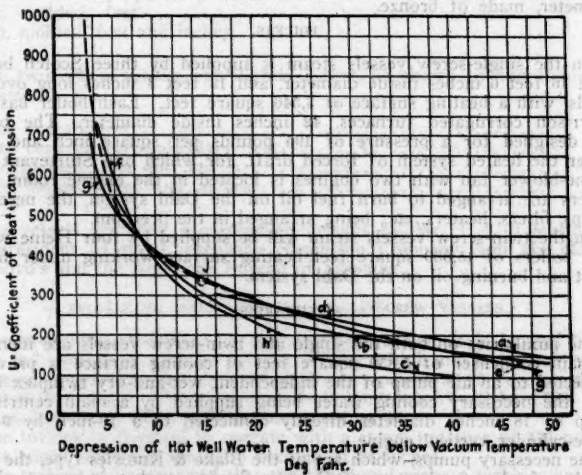


Fig. 1.—Relation Between Coefficient of Heat Transfer and Temperature Depression.

Presented at the Annual Meeting of The American Society of Mechanical Engineers, New York, December, 1915.

zone of active condensation, that is, on how much of the surface does work and how much is idle. As will be shown, the problem is one of hydrodynamics as well as of heat transmission.

These variations being commonly attributed to the presence of air and to imperfect stream distribution and penetration, modern high-vacuum surface condensers are equipped with large-capacity air pumps and are designed with liberal areas for the flow of steam through the tube bank, with the minimum depth of tube bank between inlet and outlet. When careful attention is also given to excluding air in leakage, high vacuums are successfully maintained. However, a condenser which will give a high coefficient of heat transfer when the test readings are taken under summer conditions with circulating water at temperatures of 70 to 80 degrees F., will also give relatively low values in winter with cold water; the warmer the water and the lower the vacuum, the smaller the mean temperature head required to transmit practically constant amount of heat through the surface.

DEPRESSION OF AIR-PUMP SUCTION TEMPERATURE AS AN INDEX OF SURFACE EFFICIENCY.

Examination of the results of a number of tests on commercial condensers, a vacuum evaporator and a laboratory condenser indicates a definite relation between the coefficient of heat transfer and the difference between the steam temperature at the condenser inlet and the air temperature at the outlet.

The water velocity, mean water temperature and tube diameter affect the rate of transmission from the tube wall to the water, that is, the transmission on the *water side* of the tube, and their individual influences have been determined in experimental single-tube condensers, so that corrections for these variables can be applied with a reasonable degree of certainty. Geo. A. Orrok's results have been used for the present purpose so far as possible.

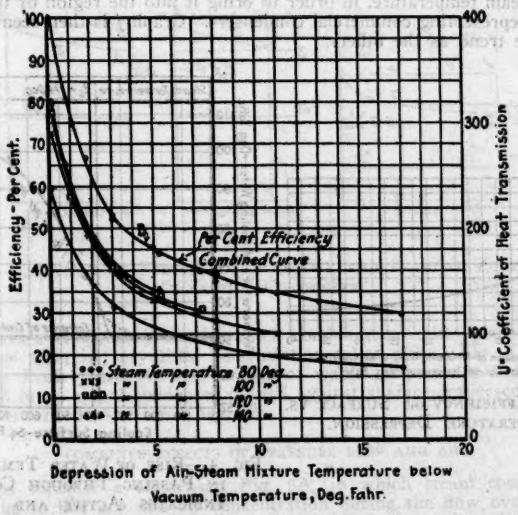


FIG. 2.—RELATION BETWEEN COEFFICIENT OF HEAT TRANSFER AND TEMPERATURE DEPRESSION.

The variables affecting heat transmission on the *steam side of the tube in an actual condenser* are complex and difficult of isolation; yet as an observed fact, their influence really dominates performance, since the coefficients of heat transfer still vary several hundred per cent. after all variables affecting heat transfer on the water side have been allowed for.

In Fig 1 the coefficient of heat transmission for a number of surface condensers is plotted against difference between steam and condensate temperatures. Graph *a* refers to a 24,000-square foot condenser loaded to 5.9 pounds per square foot; graph *b* to an 18,000-square foot condenser loaded to 7.5 pounds per square foot; graph *c* to an 18,000-square foot condenser loaded to 4.07 pounds per square foot; graph *d* to an 18,000 square foot condenser loaded to 8.09 pounds per square foot, and graph *e* to a 21,000-square foot condenser loaded to 5.95 pounds per square foot. Graphs *a* and *d* coincide. In graph *f*, for a 25,000-square foot base condenser, the coefficient rises to a value of 850 B.t.u. for a single set of observations. Graph *g* refers to five condensers of different sizes, loads and vacuums, two with wet-vacuum pumps and the others with hydraulic air pumps. Graph *h* shows the relation for a vacuum evaporator (in this case the actual air-steam mixture was observed, not the condensate).

Fig. 2 shows similar graphs based on results obtained by Prof. J. A. Smith with experimental laboratory apparatus. The four lower curves show the relations existing between the coefficient of heat transmission and the number of degrees which the temperature of the air-steam mixture is below the steam temperature corresponding to the total pressure. Calling the heat transfer at zero air 100 per cent., all the results for different steam temperatures can be combined in a single percentage of efficiency curve, as shown by B, Fig. 2.

The maximum heat transmission, obtained by Smith with pure steam, is low as compared with that obtained by Orrok and others.

Graph B is replotted as the dotted graph *j* in Fig. 1 by assuming the maximum coefficient to be 1,000 at a condensate temperature 3 degrees below steam temperature, in order to bring it into the region of the other graphs representing commercial condensers. Graph *j* is then seen to have the same trend as the others.

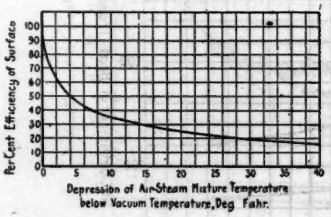


FIG. 3.—EFFICIENCY OF SURFACE VS. TEMPERATURE DEPRESSION.

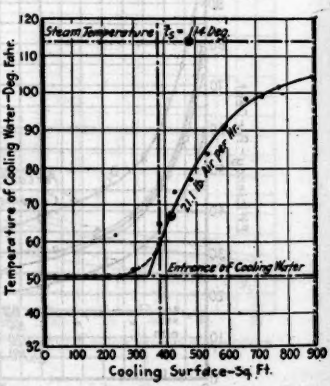


FIG. 4.—RISE OF WATER TEMPERATURE IN PASSING THROUGH CONDENSER INDICATES ACTIVE AND INACTIVE ZONES.

The graph of Fig. 3 was obtained by averaging the results represented in graphs *a*, *b*, *c*, *d*, *e*, *f*, *g* and *h*, and expressing the heat transmission as a percentage of the maximum obtainable under the conditions of rate of heat transfer through the tubes and from tubes to water, prevailing in the condensers considered.

The performance of a condenser is usually dominated, to the exclusion of other factors, by the extent of the active and inactive zones of condensation. Thus tests show that the coefficient averaged for all the surface may *increase* as the water velocity *decreases*. The explanation is to be found in the lower vacuum and smaller specific volume of steam corresponding to the lower velocity, smaller quantity and higher final temperature of circulating water, under which conditions there is greater steam penetration and more surface is brought into action.

Other explanations of the variations in heat-transmission coefficient have been founded principally on the variables affecting the water side of the tube or on the theory that a constant coefficient would be obtained by assuming the total heat transfer to vary as some fractional power of the instantaneous temperature difference instead of the first power.

We may assume for the present that the coefficient of heat transfer for a single tube immersed in steam depends on the water velocity (in an open tube or manner of agitation in other tubes), size of tube, material and cleanliness of tube and mean water temperature, but that all of these factors taken together do not explain the wide disparity of results obtained with actual condensers. We may also assume in the light of evidence that the total heat transfer is a function of the first power of the temperature difference and not of a fractional power as suggested by Loeb and Orrok.

Measurements of rise in temperature of circulating water in multiple-pass condensers have shown that there are two fairly well defined zones

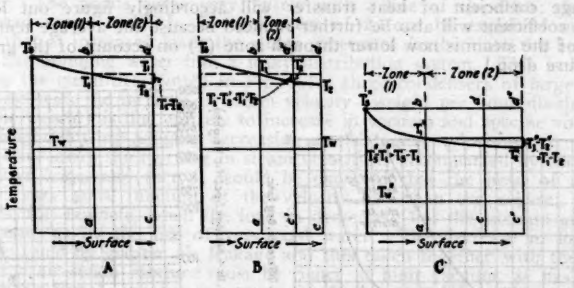


FIG. 5.—VARIATION OF PROPORTION OF ACTIVE AND INACTIVE ZONES IN AN "ELASTIC" CONDENSER.

in a surface condenser; one in which condensation takes place actively, the other wherein very little condensation takes place. Fig. 4, from a paper by Prof. Josse, shows readings of water temperature and of heat absorbed in an actual condenser plotted against surface traversed. About 40 per cent. of the surface is inactive.

COMBINED EFFECTS OF PRESSURE DROP AND AIR.

Let us consider a condenser, Fig. 5A, in which steam condensation occurs up to line *a b* and air concentration during the flow over the remainder of the tube surface; *i.e.*, nearly all the heat is transmitted to the

water in that part of the surface preceding the line *a b*. The water temperature is assumed constant at T_w. In the first zone the temperature drop to T_1 is almost entirely due to the pressure drop required to overcome the pneumatic resistance offered by the condenser structure to the flow of a large volume of steam. From this region on the temperature drop corresponds to the reduction in partial pressure of the water vapor, the drop in total pressure being negligible, due to the fact that the volume flowing in this zone of the condenser, where only 1 or 2 per cent. of the steam remains, is very small. The partial pressure of the air at exit is represented by the difference of the vapor pressures corresponding to the temperatures T_1 and T_2, T_2 being the final air-pump suction temperature.

Now assume that the air leakage is reduced, or the air-pump capacity increased, so that the air withdrawal occurs at some lower partial air pressure represented by the smaller difference T_1—T_2' (Fig. 5B). A part of the zone (2) may then be dispensed with and, as the same initial steam temperature T_1 is maintained with less surface, the average coefficient of heat transfer will figure out higher.

Again, assume the case in Fig. 5C, wherein the water temperature is reduced and a higher vacuum maintained. If all of zone (1) up to *a b* is to remain active, the pressure drop expressed in head of steam must increase approximately in proportion to the square of the steam velocity and specific volume. Temperature T_1'', is therefore closer to the water temperature T_w'' than before. Furthermore, the difference between this temperature T_1'' and the air-pump suction temperature T_2'' must increase in order to give the same partial air pressure with a lower total pressure. This is explained by Fig. 6. For a constant partial air pressure, which means constant air leakage with constant pump displacement, the temperature difference and the percentage of air richness must both increase rapidly as the vacuum increases. For these two reasons more surface will be required in zone (2) of Fig. 5C extending it to *c''*—*d''* and the average coefficient of heat transfer will accordingly figure out lower. [The coefficient will also be further reduced because the average temperature of the steam is now lower through zone (1) on account of the greater pressure drop.]

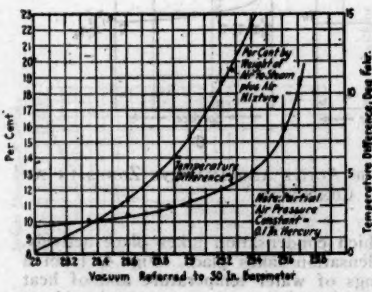

Per Cent
Temperature Difference, Deg. Fahr.
Per Cent by Weight of Air in Steam and Air Mixture
Temperature Difference
Note: Partial Air Pressure Constant = 0.1 In. Mercury
Vacuum Referred to 30 In. Barometer

FIG. 6.—PERCENTAGE AIR IN MIXTURE AND DIFFERENCE OF TEMPERATURES CORRESPONDING TO TOTAL PRESSURE AND THAT ACTUALLY EXISTING, FOR CONSTANT AIR PRESSURE OF 1/10 INCH.

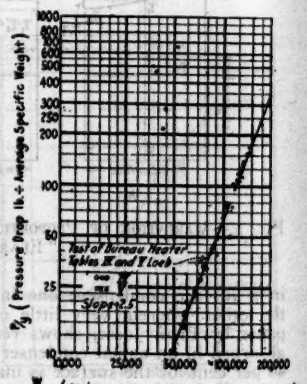


FIG. 7.—RELATION BETWEEN PRESSURE DROP AND STEAM VELOCITY.

Many variables have to be taken into account to calculate the actual relative depths of these zones in a condenser. Among the independent variables which would need to be considered may be mentioned water velocity, initial water temperature, weight of water, tube diameter and cleanliness, arrangement of water passes, steam temperature, amount of air, air-pump capacity, amount of surface and, perhaps most important of all, the arrangement or geometrical configuration of heat transmitting surface. However, we find that if the difference between vacuum temperature and air outlet temperature $T_1 - T_2$ is increased by increasing pressure drop or by increasing the amount of air, there is a decrease in the average coefficient of transmission; and conversely if $T_1 - T_2$ is decreased by decreasing pressure drop or decreasing the amount of air, there is an increase in the average coefficient. Empirical results showing the same relation have been given in Figs. 1, 2 and 3.

PRESSURE DROP.

If the proportion of steam condensed per tube, the depth of penetration and the steam density were all constant, the loss of head through the first zone of the condenser would vary as the square of the velocity, or according to the well-known relation for inelastic fluids $h = v^2/2g$. If the amount of steam condensed per tube and the steam density remained constant, but the penetration increased in proportion to the velocity of entrance, the pressure drop would vary as the cube of the entrance velocity.

Pressure drops actually observed usually fall between these limits. By dividing both the steam weight flowing and the pressure drop resulting by the mean specific weight of the steam, test readings can be expressed in terms of steam velocity and "head" of steam.

In Fig. 7 are plotted on logarithmic paper the heads and proportional velocities obtained in the experiments by Loeb on the Bureau feed-water heater. The points agree very closely with a line of slope 2.5; that is, the head varies as $V^{2.5}$. The exponent of 2.5 can be explained on the basis that at increased loads the steam penetrates farther into the heater; it is like pumping water into a leaky distribution system. This is borne out by the quantities plotted in Fig. 8 for three condensers of large size. In these tests the increase in steam velocity (weight per hour divided by specific weight) is due entirely to increase in vacuum and specific volume, the weight flowing actually decreasing at the higher velocities.

In some cases the increase in steam velocity is accompanied by decrease in active zone, and then it would be expected that the head of steam would vary as a function of the velocity less than the square. This occurs, for example, when the load in decreased, but the vacuum greatly increased, as by the use of very cold water. The decrease in load is accompanied by greater air leakage and this taken together with the fact that the air-steam mixture must be richer at high vacuum, as has been shown by Fig. 5, causes an increase in the inactive zone and shortening of the active zone. At the same time the velocity of the steam increases due to the higher vacuum.

If the depth of the active condensing zone is assumed to remain constant, however, we can assume that the head lost varies at least as the square of the velocity. In estimating performance this is a provisional basis of calculation; for, as will now be evident, to calculate the pressure drop it is necessary to know the depth of penetration, which in turn depends on the average heat transfer which is the final result sought by calculation.

CALCULATION OF SURFACE EFFICIENCY.

Assume that the several characteristics of a surface condenser are known for 28.5 in. vacuum, under which conditions the pressure drop through the condenser is 1/10 in. of mercury column and the air-pump

capacity and air inleakage are such that the partial pressure of the air is 0.124 in. of mercury, so that the depression of the air-pump suction temperature is 5 degrees below the steam temperature, corresponding to an efficiency, from Fig. 3, of 45 per cent.

We wish now to know the conditions at 29 in. vacuum. We have seen from Fig. 5 that additional cooling surface is required in zone (2) in order to reduce terminal pressure, so that the weight of steam at the greater volume corresponding to the higher vacuum can penetrate to the same depth in zone (1), wherein, in other words, greater pressure drop will occur. In the actual condenser, the surface remains constant, so that there must be a re-adjustment of working conditions, zone (1) becoming smaller because of resistance to the flow of the greater steam volume, also because more surface is required in zone (2) for air concentration.

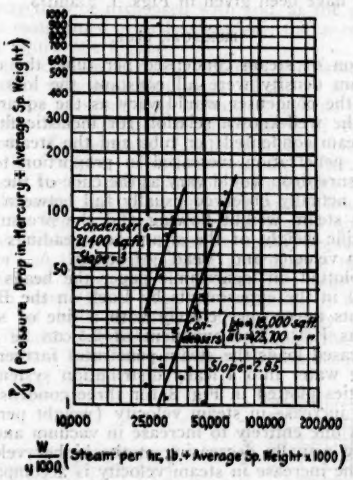


FIG. 8.—RELATION BETWEEN PRESSURE DROP AND STEAM VELOCITY.

[We must bear in mind that zone (1) is not only affected by pneumatic conditions but by the performance of zone (2) and of the air pump. Thus if we slow down the air pump, zone (1) will be shortened with consequent decrease in vacuum.]

We can calculate the pressure drop for the assumed elastic condenser of Fig. 5, wherein zone (1) always remains the same. This, in connection with the depression due to air pressure, leads to a temperature difference $T_s - T_2$ and a corresponding surface efficiency, which we will apply to the actual condenser.

Thus the pressure drop at 28.5 in. of 0.1 in. becomes 0.148 in. for the larger volume at 29 in. vacuum and the total pressure at the air suction is (1—0.148)=0.852.

Modern air pumps, particularly the hydraulic types, running at constant speed, remove practically a constant volume of air, that is, have constant displacement or equivalent displacement under wide conditions of operation. Therefore with constant air inleakage the partial pressure of the air in the air-steam mixture withdrawn from the condenser must also be a constant, neglecting the slight effect of temperature on air volume.

The partial air pressure thus remains equal to 0.124 and the partial vapor pressure is 0.852—0.124=0.728 in. This pressure corresponds to a steam temperature of 69.5 degrees, which is a depression of 9.5 degrees below the vacuum temperature, so that the efficiency of the condensing surface is now 36½ per cent., according to Fig. 3.

Similarly the depression for 29.2 in. vacuum is found to be 13.8 degrees and the efficiency 32 per cent., and so on for other vacuums, and a graph can be plotted as shown in Fig. 9, graph *A*.

Graphs *B*, *C* and *D* were obtained similarly, *B* being for a lower partial air pressure, due to less air leakage or a larger capacity pump, *C* for the same air conditions as *A*, but less pressure drop (as with better spacing of tubes), and *D* for both reduced air and less pressure drop.

Graph *B:* Reducing the air leakage (or increasing pump capacity) gives less temperature depression and hence raises the efficiency particularly at moderate vacuums.

Graph *C:* Reducing the pressure drop raises the high vacuum end of the efficiency curve and also extends it to higher vacuums. The efficiency of the surface at 29.4 in. vacuum is 36 per cent. as against 25 per cent. in graph *A*.

Graph *D:* Reducing both air leakage and pressure drop results in large increase in efficiency as shown by this graph.

ZONE CONDENSERS.

We have been considering the conditions on the steam or steam-air side of the tubes, and the variations in the relative extension of the active zone of the condenser. The performance under any conditions will also be affected by the conductivity of the heating surface itself, dependent upon its cleanliness and also upon the conductivity from the metal to the water within the tube. Fouling or scaling of condenser tubes will reduce heat transmission and can be corrected by cleaning only.

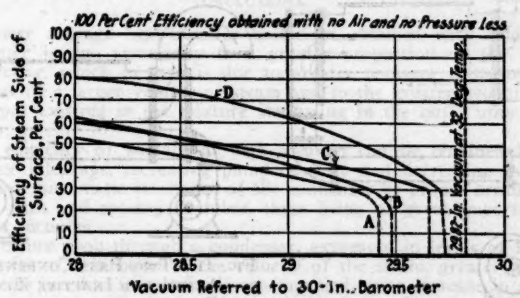


FIG. 9.—EFFECT OF AIR AND PRESSURE LOSS ON EFFICIENCY.

Assuming a clean condenser, the conductivity from the tube to the water becomes the controlling factor in the active zone, and therefore by increasing velocity, greater heat transmission and better vacuum can be obtained. In the inactive zone, however, any increase in velocity of the water is of little benefit. There is only a very small amount of heat being transmitted, so that the difference in temperature between the metal tube and the water is very small compared to the difference in temperature between the metal wall and the surrounding stagnant steam-air mixture. Hence any reduction in the already small temperature difference from metal to water gained by increase in water velocity has an imperceptible effect on the overall temperature difference between steam-

air mixture and water and the corresponding rate of heat transfer, just as an increase in the size of the valve placed on the end of a long, small diameter pipe line, will not result in any appreciable increase in quantity of water discharged.

Thus whereas it is profitable to increase the water velocity through the tubes in the active zone of the condenser, any such increase is of no value commensurate with its cost in the inactive zone. A decrease of velocity in these tubes is even desirable, because by such decrease the power for pumping the circulating water can be decreased, or else for the same power a higher velocity of water can be maintained through the tubes of the active zone where it will do some good. A large flow of water is not required for heat absorption in the inactive zone, as relatively little heat is transmitted here.

The application of this principle is shown by the sketches of Figs. 10, 11 and 12. In Fig. 10 a condenser is arranged with two passes, one containing three-fifths of the surface and the other two-fifths of the surface, the former being intended to comprise the active and the latter the inactive zone. All of the water is passed through the tubes of the active zone at high velocity. A part only of the water passes through the tubes of the inactive zone, the rest being short-circuited through the by-pass. If the velocity of the water through the inactive-zone tubes were one-fifth of that through the active-zone tubes, the head lost would only be 1/25 as great, with corresponding saving of power for pumping the circulating water. Thus, for the same circulating pump power as ordinarily required with a standard two-pass condenser, more water may be pumped and higher water velocity maintained through the active tubes

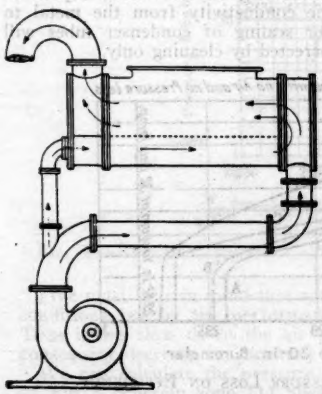


FIG. 10.—TWO-PASS CONDENSER WITH ACTIVE AND INACTIVE ZONES.

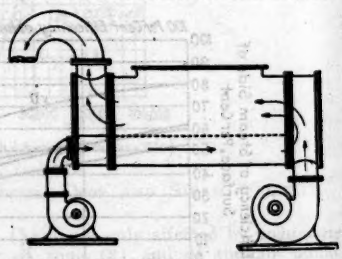


FIG. 11.—TWO-PASS CONDENSER WITH ACTIVE AND INACTIVE ZONES, TWO PUMPS.

of the condenser. Fig. 11 shows how the same conditions are obtained using two circulating pumps.

In Fig. 12 is shown a three-pass condenser in which the velocity in the lower zone is small, in the middle zone fairly high, and in the upper zone the maximum. All of the water goes through the upper zone, but only part of the total through the other zones. By manipulation of the valve the quantity of water going through the zones is varied to take care of different conditions of operation in summer and winter, and of

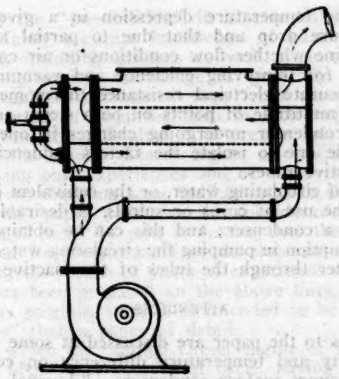


FIG. 12. — THREE-PASS CONDENSER WITH THREE ZONES OF DIFFERENT ACTIVITIES.

load, air leakage, air-pump capacity and vacuum. A three-pass zone condenser may also be built with, say, half the tubes in the bottom or first pass and the other half of the tubes divided between the two upper passes. The objection to this design is that the inactive zone is too large (one-half the total) and the water velocity still too high.

CONCLUSIONS.

The lower average heat-transfer coefficient obtained with cold circulating water is due principally to a greater proportion of the surface being inactive, which in turn is due to greater pressure drop incidental to the flow of a larger volume of steam and to the greater proportion of air to vapor existing in the mixture stagnating in the outlet zone of the condenser.

Better efficiency of surface at high vacuum can be obtained by preventing air inleakage, increasing pump capacity and decreasing the pressure drop or pneumatic resistance of the condenser by proper arrangement of steam path and spacing of tubes, these being of great importance at very high vacuums.

The pressure drop through a condenser, expressed in terms of head of steam, varies as a function of the velocity of the steam, greater than the square if the increase in velocity is accompanied by an increase in average heat transmission coefficient and in depth of penetration, or smaller than the square if the increase in velocity is accompanied by a decrease in average heat transmission coefficient and in depth of penetration.

In purchasing high-vacuum condensers comparison should be made of the pneumatic resistance of the structures on the basis of velocity of flow at each row of tubes and the number of rows in series through which steam must flow. Attention should also be given to possibilities for the formation of air pockets out of the line of flow, considering both transverse and longitudinal sections.

The highest coefficients of heat transmission are obtainable in condensers of moderate size in which the smaller depth of tube bank lessens the pneumatic resistance.

The depression of the air-suction temperature below the inlet-vacuum temperature is an index of the surface efficiency on the steam side.

By analyzing the temperature depression in a given condenser into that due to pressure drop and that due to partial air pressure, it is possible to determine whether flow conditions or air conditions offer the greater possibility for improving efficiency and vacuum.

By means of accurate electrical resistance thermometers temperatures can be taken at a multitude of points on both steam and water sides of the surface of a condenser undergoing changes in operating conditions, which would enable one to isolate the factors influencing the extent of the active and inactive zones.

A high velocity of circulating water, or the equivalent increase in water-film agitation by the use of cores or spirals, is desirable in the tubes of the active zone of a condenser; and this can be obtained, without additional power consumption in pumping the circulating water, by reducing the velocity of the water through the tubes of the inactive zone.

APPENDICES.

In two appendices to the paper are discussed at some length the effects of viscosity, velocity and temperature difference on coefficient of heat transfer in high vacuum surface condensers.—"Journal Am. Soc. M. E."

THE PRESENT POSITION OF THE MARINE DIESEL ENGINE AND ITS POSSIBILITIES.*

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Present circumstances render it difficult to make a correct survey of the strictly *present* position of the marine Diesel engine, although this does not affect to the same extent the degree of accuracy—or inaccuracy—of deductions to be drawn, when considering the possibilities of the future. With a very few exceptions the processes of evolution and development of the marine Diesel engine have, unfortunately, been carried out entirely on the Continent, and it is there that the largest experience has been gained and most progress made, to the detriment of progress in this country. One regrets to have to admit this. It is hoped that this state of affairs will soon be righted after the conclusion of the war, and there is little doubt as to this, for the British have a way of doing things thoroughly, although they may be rather late in making a start.

The marine Diesel engine is essentially an engine of detail. The failure of one or more essential details may involve very serious consequences. Also incorrect deductions may be drawn from failures of detail. For success and real progress there must be the most complete collaboration between the designers, the works and the testing departments of any one firm of makers, and a frank and frequent interchange of ideas, through proper channels, between the various builders and users. Owing to the war, communications—particularly technical ones—have been rendered difficult with neutral, and necessarily impossible with enemy countries. Also both ourselves and the enemy have for many months had to devote our whole attention to other things than the further development of the marine Diesel engine, and owing to the same cause there has been indirectly a retardation of progress in neutral countries. Therefore it may be taken, with reasonable accuracy, that the present position closely agrees with that of eighteen months ago.

*Institution of Naval Architects, April 13th, 1916.

In considering the "possibilities," it is essential that full advantage be taken of a fairly accurate knowledge of past experiences and of progress to date, including failures as well as successes. In fact, the most valuable engineering knowledge is generally obtained from the experience of troubles surmounted than from cases of their complete absence. The process of acquiring such valuable experience is generally costly. Apart from the difficulties, the present lull in the progress of development of the marine Diesel engine affords a convenient opportunity of reviewing past experiences and of surveying the present situation with a view to obtaining some indication as to the future. Although the subject of this paper precludes the consideration of the land type of Diesel engine, it must be said in justice that the marine type would not have attained its present state of development so soon, had it not had the benefit of the results of years of experience with land types. The present paper has been prepared on the above lines, and is intentionally made as brief as possible, since it is intended to be more of the nature of an "indicator" than a paper of detail.

The cycle on which the original Diesel engine of 1892 was intended to work was an "ideal" and theoretical cycle, having the highest thermal efficiency attainable with the large range of temperature proposed for the cycle. Economy was the keynote of the principle. Compression of air—air only, not a mixture of air, with gas or vapor—was for the first part "isothermal"—due to the injection of cold water into the cylinder—and, after the supply of injection water was cut off, the remaining portion of the compression stroke was "adiabatic"—due to the lagging of the uncooled cylinder—with consequent rapid rise in temperature. At about dead center, fuel—at first coal dust—was injected into the hot air contained in the cylinder at such a controlled rate that combustion, which commenced at dead center, continued during the period of injection at such a rate that the products of combustion expanded "isothermally." At a certain point in the stroke the supply of fuel was cut off and the gaseous contents of the cylinder expanded adiabatically until at the end of the stroke they had expanded to and were exhausted at atmospheric temperature and pressure.

This theoretical cycle was very soon found too ideal to be a commercial proposition, and practical reasons led at once to a reduction in the ratio of compression, to the abandoning of water injection, to the water jacketing of the cylinder, to the admission and regulation of the supply of fuel so as to give combustion at constant pressure, and to the exhausting of the products of combustion at a pressure and temperature both considerably in excess of atmospheric. The immediate results were a reduction of the maximum pressure in the cylinder and of stresses consequent thereon, a reduction of range of working compression temperature—still much more than sufficient to ignite any of the fuels used then or since—but they involved a large increase of temperature during combustion, and a considerable falling off of thermal efficiency. There was, however, an increased output of power from a given size of cylinder, or, what is the same thing, a reduction of size for a given output. The use of coal dust was abandoned and liquid hydrocarbons adopted instead.

All Diesel engines of the present time aim at combustion at approximately constant pressure, are water jacketed round the power cylinders and exhaust at considerable pressure—say, up to 40 pounds per square inch by gage—and at comparatively high temperature—say, 500 degrees F. and above. In regard to the processes happening in the lower cylinders, there has been practically no change during the last fifteen years. The first marine Diesel engines worked on the "four-stroke" cycle, and there are numerically many more marine Diesel engines in operation working on this cycle than on the later "two-stroke" cycle, which only dates from 1906. This is but a natural outcome of development, and the two types,

viz.: two and four-stroke cycles, will be compared later as we pass points in the paper. The earlier marine Diesel engines did not extend the cooling to the interiors of the piston crowns, being small in power; but, as sizes increased, it was found necessary internally to cool the piston with air, water or oil—particularly so in two-stroke engines—to water-cool the exhaust valves, where fitted, and in some cases their seats.

As far as it has been possible to ascertain, the total number of Diesel-engined motor vessels completed up to the end of December last is about seventy, having a total deadweight capacity of about 350,000 tons, and a total brake horsepower of about 110,000. These vessels are nearly all of the cargo-carrying mercantile type; war vessels and submarines are not included. It is also understood that about a further fifty vessels of over 300,000 tons deadweight capacity total, and of nearly 100,000 brake horsepower total, were on order at the end of last year, and several large vessels of comparatively high power have been ordered during the past few weeks. When it is recalled that in 1911 the total number of motor vessels completed during that year was but three, of 5,000 tons deadweight capacity total, and 1,200 total brake horsepower, it will be seen that progress has been extremely rapid in this type of vessel.

The engines fitted to these vessels are practically all of the single-acting long-stroke slow-running mercantile type, differing but little from the land type, except that they are reversible, that they mostly have cross-heads, and that little restriction has been placed on weight or space occupied. The later motor ships have engines, the powers of which range from about 750 to 3,600 indicated horsepower per shaft, *i. e.*, from 125 to 600 indicated horsepower per cylinder. The fuel economy is remarkably good, the consumption varying from .3 pound to .33 pound per indicated horsepower per hour, or one-fourth to one-fifth that of good reciprocating steam machinery of corresponding size and type. The weight of the propelling machinery and selfcontained auxiliaries varies from 3 cwt. to 5 cwt. per indicated horsepower. A few of the larger engines work on the two-stroke principle, single-acting almost without exception and develop up to 2,500 brake horsepower—say, 3,600 indicated horsepower—per shaft, or 420 brake horsepower—600 indicated horsepower—per cylinder. The fuel consumption for this type is from 5 to 10 per cent. higher per unit of brake horsepower—due to the work required for scavenging—but per unit of indicated horsepower it is practically the same as for corresponding four-stroke engines. The weight per horsepower is about 25 per cent. to 40 per cent. less than for four-stroke engines; the space occupied is considerably less, especially as regards height. They are eminently more suitable to the higher powers than the four-stroke engine. At least one of the vessels is fitted with two-stroke double-acting engines of about 850 brake horsepower per shaft, or 280 brake horsepower per cylinder. All the preceding figures relate to slow-running engines of the mecantile type.

Apart from the above, there are at least three large experimental engines installed for testing on shore, designed for 2,000 to 2,500 brake horsepower per cylinder, of slow-running type, and suitable for propulsion; but, although each engine has developed its designed power, the troubles experienced in two cases at least have been very great, and it is doubtful whether Diesel engines having such large powers per cylinder will ever be remunerative to their makers or sufficiently reliable on service to justify their adoption on a large scale either for mercantile or naval purposes. Generally, for naval work, engines having a relatively high speed of rotation, of light weight, and occupying as little space as possible are required; and in many special cases these conditions are so vital that failure to comply prohibits their application. It is also desirable that the engines be made as robust as possible, and that reliability and fuel economy shall not be sacrificed unduly.

The first high-speed naval-type engine was completed about 1906, and since that time its development has been even more rapid than that of the slow-running mercantile type of engine previously referred to. The maximum powers per cylinder in high-speed engines at present at work are not so great as those of the mercantile engines, but the former are rapidly approaching, and may soon overhaul, the latter.

One is not permitted to state the number of vessels equipped with such engines, or to furnish more than the barest information about the engines themselves. Suffice to say, that there have been constructed and set to work some hundreds of such engines, aggregating over 500,000 brake horsepower, the powers per shaft being generally slightly under 1,000 brake horsepower and the powers per cylinder varying from 100 to 150 brake horsepower. The speeds of rotation vary from 350 to 500 revolutions per minute. There are a few instances where the power is as high as 1,500 brake horsepower per shaft, the power per cylinder varying from 125 to 250 brake horsepower. It is also known that in the summer of 1914 certain engines of submarine type, designed to develop 2,500 brake horsepower per shaft, or over 300 brake horsepower per cylinder, were nearing completion, and they are now doubtless on service. Quite recently an experimental single-cylinder submarine-type engine was tested on shore, and developed rather more than 500 brake horsepower at about 350 revolutions per minute. The reliability was said to be extremely good, as determined by a shop test of considerable duration, as was also the fuel economy. An eight-cylinder engine made up of such units would therefore give about 4,000 brake horsepower per shaft, which exceeds the power per shaft of the largest mercantile engine at present built and installed. All these high-speed engines are of the single-acting vertical type, and the majority are "forced-lubricated." Rather more than half of them work on the four-stroke cycle, and the bulk of these latter are not reversible. As might be expected, those having the larger powers per cylinder work on the two-stroke cycle, and are directly reversible.

The weights per brake horsepower of this type of engine are approximately as follows:—For engines for continuous work, such as driving dynamos, 112 pounds per brake horsepower. For propulsion, 84 pounds per brake horsepower for the smaller to 56 pounds in the larger units per cylinder. These weights refer to the engines themselves, including such accessories as lubrication and circulating pumps, air compressors and reservoirs necessary to start and run them. In the small and medium sizes of high-speed engines, say, up to 800 brake horsepower per shaft, the saving in weight by the adoption of the two-stroke cycle is not usually very great, say, 15 to 20 per cent., but in the larger sizes the saving rises rapidly. The fuel consumption per indicated horsepower per hour is generally somewhat higher than for slow-running engines, say, about 5 to 10 per cent., due principally to higher velocities through valves and ports, and more rapidly "functioning," but this is not necessarily so. There is about the same disparity between the fuel consumptions of two and four-stroke engines of high-speed type as obtains between these cycles in slow-speed engines. Owing to limitations of space, principally of head room, this type of engine has hitherto been made single-acting only, and no crossheads or slipper guides, in the ordinary accepted sense, have been fitted, although in certain designs the walls of the power cyclinder and their pistons, have been relieved of the stresses due to obliquity of the connecting rods, by fitting combined scavenging and guide pistons beneath them. Naturally the wear and tear is greater in high than in slow-speed engines, but this is largely counteracted, at least as far as bearings are concerned, by the generally efficient systems of forced lubrication adopted in most high-speed Diesel engines.

In order that any Diesel engine shall render reliable service, it is neces-

sary that certain essentials in design, construction, attendance and maintenance shall be strictly complied with. As before stated, the Diesel engine is an engine of detail. In designing Diesel engine details the particular function which the part is intended to perform must be most carefully considered in conjunction with the conditions under which it has to operate.

Essential factors to be taken into account, which do not, of course, all apply to every detail, are:

(*a*) *In Designing.*

(1) Adequate strength and stiffness.
(2) Suitability of material.
(3) Accessibility and ease of adjustment.
(4) Satisfactory arrangements for the dissipation of heat and adequate cooling.
(5) Absence of distortion under working conditions, during erection, and when replacing after examination or overhaul.
(6) Sufficient, but not excessive, working clearances.
(7) The results of experience on service should be rendered available for use by the designing staff.

(*b*) *During Construction.*

(1) The use of suitable materials, particularly in castings subjected to heart or / and pressure, and those intended to contain fuel.
(2) Accuracy in machining and in erection to ensure true form, correct fits and clearances, and preservation of alignment.
(3) Absolute correctness of the timings, openings and periods of all valves controlling air or fuel.

(*c*) *Attendance when Working and Maintenance.*

(1) It is most desirable that the engineers in charge and their principal assistants be afforded an opportunity, during construction, of becoming acquainted with the details of the machinery which they will later have to handle, and with the working of the machinery during trials on shore.
(2) Fully detailed arrangement drawings and carefully drawn-up instructions—both for working and maintenance—are of great assistance in ensuring proper management, if followed.
(3) Tampering with the adjustments, or alteration of the design of important details should be discouraged and prevented, at least until an expression of expert opinion has been obtained.

The principal reason for referring to the above points is that owing to lack of attention to the points in question, each one of them has in some case or other—and sometimes in many cases—caused trouble or accident to the machinery when actually on service.

A few of the details which have given trouble and which need special care and attention will now be considered, with a view to facilitating the elimination of such troubles and to point the way to further developments.

(1) One of the principal difficulties has been the selection of proper materials for cylinders, pistons and cylinder covers. As usually designed these parts must be castings. Many such intricate and costly castings in iron have failed owing to stress, irregular expansion or inadequate jacket cooling. A case is on record where over a hundred cylinder castings were made before four good castings were obtained. This is, of course, a very exceptional case. Cast steel has been tried and has failed; "semi-

steel," *i. e.*, cast iron containing a percentage of cast or wrought steel, is but little better; manganese bronze is being tried in some cases for covers, but its success is very doubtful; and the best material so far is a carefully selected grading of cast iron, having a very close grain, a high tensile strength, a small coefficient of heat expansion, and at the same time sufficiently fluid completely to fill the mold without undue internal stressing on solidification. It is doubtful whether the annealing of cast iron is of any real benefit. Slow solidification and avoidance of chilling seems preferable, judging by personal experience. The difficulties above mentioned increase with increase of size, and any simplification in the design would be really helpful.

(2) Frequent sources of trouble, which generally are discovered before an engine leaves its works, are directly traceable to inaccuracies of workmanship. Holes and bores which should be circular and parallel and made to certain dimensions are not so. Two such holes which in the engine are contiguous, and which should be concentric and coaxial, are not so. Efforts made to correct these inaccuracies of manufacture lead to still worse troubles; valves and spindles are bent, and may seize, the valves may remain open when they should be closed, and when in the closed position they may leak. This applies particularly to fuel valves, fuel-pump plungers and air-compressor valves. Such cases aggravated by faulty alignment have also occurred with disastrous results in the main motion parts, *i. e.*, pistons, cylinders, and crosshead-pin bearings. Crank shafts have been received for fitting whose pins and journals were oval by more than the diametral working clearance permissible, and conical at that, whilst the crank pins, instead of being in strict parallel alignment with the journals, have not only been inaccurate in two planes, but have been so disposed that the deviations in angular spacing have exceeded the limits of adjustment permissible on cams and valve gearing. Errors of workmanship which may be relatively unimportant under the low maximum pressures and temperatures obtaining in steam machinery are of the utmost importance in Diesel engine work.

Passing now to the "running" conditions—

(3) The most frequent source of trouble at starting is "loss of air," and the obvious remedy is an efficient means of economizing its use, a sufficient supply and adequate auxiliary capacity for replenishment. Any economy, either in injection or starting air, is welcome, and skill, care and experience in manipulation of the engine are the most important methods of securing this economy; particularly as regards starting air. Sluggish starting may be due to insufficient compression owing to: (*a*) piston ring leakages, (*b*) valve leakages, (*c*) wearing down of bearings, or (*d*) failure of fuel pumps to "pick up" their fuel promptly, due to air locking, leaky valves or incorrect "setting" as regards quantities. Each of these causes is readily capable of discovery and remedy if the proper steps are taken.

(4) If the starting valves are not operating as they should, or if there is undue resistance to rotation of the propeller shaft—grounding, or a foul hawser, are not unlikely causes—the engine may fail to start, and it is necessary to wait until the air supply is replenished before a further attempt is possible; with air starting this is inevitable, although "cross-connecting," with adequate reserve storage, will minimize the trouble. An invaluable aid is the inter-connecting of the air-injection vessels of all the engines in the same compartment, since without injection air a Diesel engine cannot be started.

(5) One of the most serious troubles is that which may arise from the introduction into the working cylinders of fuel before the proper time. In such a case the engine ceases to be a Diesel engine, and pre-ignites—mostly after miss-firing, and therefore with the chances of a multiple charge—in which case combustion takes place not at constant pres-

sure as desired, nor at constant volume as in ordinary types of internal-combustion engines, but in a volume which is being reduced during combustion. This means large increases of pressure, temperatures and stress during combustion, which may result in violent knocking, overheating and stressing of parts, and even in disaster. The most frequent cause of this trouble is that of stuck-up, sluggish or inoperative fuel valves—due either to bad alignment or to seizure in the glands. Improvements in design will minimize the risk.

A further frequent cause of miss-firing is the cooling of the power cylinders due to the rapid expansion of high-pressure starting air, the admission of which may be frequently repeated in cases of faulty starting, and the corresponding admissions of fuel to the cylinder may be overlooked, to cause disaster when ignition does occur. Another, but controllable cause, is the abuse of the process of "priming"—*i. e.*, filling the fuel-delivery pipes with fuel. So long as adequate precautions are taken, the process is safe; but if, owing to a fuel valve being open, or subsequently lifted off its seat, or by-pass valves being left shut up, fuel enters the cylinder and is allowed to lie on the piston crown, accident may, and probably will, happen due to pre-ignition, as just described. Such results may be disastrous in large engines even with relief valves fitted to the combustion spaces.

(6) Air compressors have given, in the early stages of development, considerable trouble. The principal causes have been excessive lubrication, inadequate cooling during compression and between the stages, inefficient drainage and insufficient reliefs. These difficulties have now been entirely surmounted, and the latest compressors run for months with only moderate attention. In certain makes of engine, compressors have been dispensed with for the injection of fuel, but it is an open question whether the advantages balance the disadvantages, especially if air is required for reversing the engines. The mechanical pulverization of heavy fuel cannot be made so efficient as with air.

(7) Air-starting valves will occasionally stick, and in such cases either starting air is wastefully admitted during the whole of the stroke, and is supercompressed in the cylinder, and / or hot products of combustion pass from the cylinder into the starting pipe. Means should be provided for working the starting valves when the engines are under way, to ensure freedom of motion and absence of leakage.

(8) It may safely be said that after training no abnormal skill is necessary either to discover the cause of these troubles or to remedy them—it is a common-sense process mostly, but one feels that, as previously stated, full advantage should be taken of the "results of experience." It will be noted that no complaint is made as to reversing. This difficult problem has been satisfactorily solved in both four and two-cycle engines. It is true that this is more easily affected in the latter case than in the former, but efficiently in both.

We now come to the possibilities of the future, based on previously ascertained facts, and here one walks, of necessity, on delicate ground. Nevertheless, it is hoped that the conclusions drawn will at least be logical, and certainly that they will not be exaggerated, for exaggeration is worse than misleading.

There is a limit of size below which Diesel engines cannot be made at all, and, further, a limit below which they cannot be made commercially. A factory was established at Münich to manufacture Diesel engines of 5 horsepower per cylinder on the four-stroke principle, running at 600 revolutions per minute. The engines were fairly successful, but the undertaking was not successful financially. Later, in 1911, an experimental single-cylinder engine, designed to develop 10 brake horsepower at 1,000 revolutions per minute, and working on the two-stroke Diesel cycle, was constructed and tested satisfactorily, but it was not a commercial suc-

cess, since its cost exceeded by far what would be regarded as a fair market price for an engine of the power. These cases, however, are not devoid of scientific interest, as showing that Diesel engines can be made in very small units and to operate at high speeds, although in such cases they are not marketable commodities on account of high prime cost.

Before dealing with other matters it may be of interest to refer to two cases of engines installed in the same ship, one a four-cycle and the other a two-cycle Diesel engine, each running at the same speed—500 revolutions per minute—and occupying the same space. It was found that 85 brake horsepower was the limit possible with the four-cycle engine and 125 brake horsepower that with the two-cycle engine; the weight of the engines in running order was as nearly equal as possible. The ratio of power for equal weight and space is therefore about 50 per cent. in favor of the two-cycle, which is rather surprising in units so small.

Turning now to the other extreme—the large units of power—a survey of the principles involved, from which deductions are to be drawn, may not be amiss. The results of forty years of experience with the reciprocating marine steam engine have shown that piston speeds of from 750 feet to 1,000 feet per minute are rarely exceeded, and even in the latest practice with high-speed reciprocating engines, as were fitted in torpedo-boat destroyers, 1,200 feet per minute was regarded as a limit—18-inch stroke and 400 revolutions per minute. It is therefore reasonable to regard 1,200 feet per minute as the limit to be approached, but not exceeded, and 1,000 feet per minute as a reasonable limiting working figure to deal with when considering the application of Diesel engines in high powers to purposes of marine propulsion. Already Diesel engines in ships are running at piston speeds varying from 750 feet to 1,000 feet per minute, and in exceptional cases rather higher—1,100 feet per minute. Regarding, therefore, 1,000 feet per minute as a standard of comparison, and assuming that the mean effective pressure in the cylinders is 100 pounds per square inch—which represents an average of current practice—and that due to reasons connected with dissipation of heat, the ratio of stroke to bore is unity as a minimum, preferably higher—take 1.2 for the sake of comparison—the following may be deduced when investigating or comparing types and designs. Any deviation from the basis figures assumed will, of course, make corresponding alterations in the proportions of the engine, but will not affect the general question.

Taking, first, the case of light fast vessels of the destroyer type. Modern destroyers are almost invariably propelled by means of high-speed turbines, supplied with steam by water-tube boilers of large power per unit, and generally "oil-fired." Speed being the great essential for such vessels, lightness and compactness of machinery is vital. Such steam-propelling machinery with auxiliaries will weigh from 27 pounds to 35 pounds per shaft horsepower. If, instead of such machinery, it is desired to install Diesel engines the following results:

(1) They must be reciprocating engines, since the principle of the Diesel turbine is practically and theoretically impossible with our present knowledge, due principally to the fact that only a negligible portion of the products of combustion are condensible, and the great advantage of vacuum in turbines cannot therefore be realized.

(2) They must run at high speed, and head room will practically limit the type to single-acting engines.

(3) Weight limitations will necessitate the adoption of the two-stroke cycle in preference to the four-stroke, and it may be necessary to provide for mean pressures greater than 100 pounds per square inch.

(4) There must be some substantial advantages to justify the change.

Now, the lightest Diesel engines made weigh, without auxiliaries, about 56 pounds per brake horsepower, and consume, say, ½ pound fuel per

brake horsepower, whilst steam turbine installations, complete with auxiliaries, weigh, as said before, about half as much per shaft horsepower, the consumption of oil fuel being rather under 1 pound per brake horsepower. The installations can only be comparable if the weight of machinery plus fuel for a given displacement, speed and radius of action is approximately the same in both cases; and, even were that possible, the advantage would be with the steam machinery, owing to the more rapid lightening of the vessel due to the difference in fuel burnt and a corresponding augmentation in speed. A case has been worked out in some detail, and shows that the proposal is impracticable for large torpedo-boat destroyers of high power and speed, since no saving in fuel can be made to compensate for the extra weight of the machinery in such vessels. Only when the power is comparatively small, and/or the weight of fuel carried considerably more than at present, is it possible to install Diesel engines in torpedo craft to advantage, even if at all. The case of the light unarmored cruiser is practically no better than that of destroyers, the only possibility being the installation of small Diesel engines for cruising whose economy at cruising speeds might justify their installation, although they are of necessity deadweight to be uselessly carried about at full speed, unless very special arrangements are provided, which would be both heavy and costly.

Considering now armored vessels, such as large cruisers and battleships—

(1) *Large Cruisers.*—The speed and power of these vessels is necessarily high, of the order of 30 knots and 70,000 shaft horsepower. In such large reciprocating machinery the speed of rotation must be comparatively low, since, if the power per cylinder is to be high, *i. e.*, not too many cylinders fitted, the diameter of each cylinder must be large and the stroke no less —preferably more. Hence, low revolutions with limitations imposed by the piston speed. The power per cylinder and per shaft is limited by the capacity of means for manufacturing, by propeller requirements as regards efficiency, by conditions of available head room and armor protection, apart from the great "Diesel" troubles which will certainly be experienced in very large units per cylinder. Alternative cases have been worked out in considerable detail on the basis of equal powers per shaft for the development of 72,000 brake horsepower on 2, 3, 4, 6 or 8 shafts, with 6, 8, 12, 18 or 24 cylinders per shaft, the engines being two-cycle, double-acting. A few selected examples are contained in the table, the figures being in all cases estimates, based on preceding assumptions and present knowledge.

From this table, and taking it to be fairly accurate, it is apparent that the only reasonable solution is to adopt a moderate power per cylinder—say, 750 to 1,500 brake horsepower—a considerable number of cylinders per shaft and a number of shafts—not less than four, preferably six or even eight. Under these latter conditions, Diesel engines for propelling battle cruisers are possibilities, though their adoption is likely to be deferred for some years yet, permitting meanwhile a more gradual and therefore a less artificial and more sturdy growth from present sizes.

(2) *Battleships.*—In these vessels, speed, although desirable, is not so vital as in cruisers, and the power is much less. A case worked out showed that, with an equal number of shafts as for steam machinery, equal powers and speeds with double-acting two-cycle engines can be obtained as with steam, on equal machinery weights, including auxiliaries in each case, whilst no increase in displacement is necessary, the fuel capacity need not be reduced, but the radius of action, including fuel for auxiliaries, can be increased at full speed at least threefold, and at cruising speeds at least four times. Such a proposition is possible from an engineering point of view.

Number of shafts.....	2	3	4	6	8
Number of cylinders per shaft	6	8	12	12	12
B.H.P. per shaft.....	36,000	24,000	18,000	12,000	9,000
B.H.P. per cylinder.....	6,000	3,000	1,500	1,000	750
Total number of cylinders	12	24	48	72	96
Bore of cylinders, inches.	60	42½	30	24½	21½
Stroke, inches	72	51	36	30	26
Revolutions per minute...	85	120	170	200	230
Maximum load on piston, tons	630	315	158	106	81
Diameter of piston-rod, inches	19	13.5	9½	8	7
Diameter of each cross-head pin, inches.....	26½	19	13½	10½	9¾
Length of each cross-head pin, inches	26½	19	13½	10½	9¾
External diameter of crank shaft, inches	46	36.5	29½	24	21
Internal diameter of crank shaft, inches	23	18¼	14¾	12	10½
Axial length of crank webs, inches	39	30	23½	19	16
Length of crank pins, inches	61½	39	24	20	17½
Thickness of cylinders, inches	6	4¼	3	2½	2⅛
Approximate overall length of engine, feet..	112	116	122	100	87
*Working height above crank shaft, feet.....	42	30	21	18	15
*Engine projects above waterline, feet	23	11	2	—1	—4
Approximate total weight of main propelling machinery and its auxiliaries, tons	6,500	6,000	5,250	5,000	4,750

*In a special design which is being evolved, these figures may be capable of a reduction of about 15 per cent.

(3) *Submarines.*—An eminently suitable field for naval Diesel engines is the surface propulsion of submarine vessels, and it is to this purpose that this engine has been most widely applied. The powers required do not exceed those obtainable from reliable engines, the space can be provided to accommodate them, and weight is not prohibitive. Hitherto head room available has limited the application to the single-acting type, and it is extremely doubtful whether double-acting engines can ever be fitted, due to the same reason, unless the dimensions of submarines are largely increased.

Favorable installations can be made of Diesel engines for emergency lighting, dynamo and auxiliary machinery driving in large vessels, and for the propulsion of tank vessels, fleet auxiliaries and ordinary merchant vessels of moderate powers and speeds, but large merchant vessels and ocean liners fall more under the general conditions obtaining for battleships and large cruisers. Each such case, however, has to be considered on its merits and known conditions of service, so that it is not well to generalize, particularly in mercantile propositions.—"The Engineer."

AERONAUTICS.

To review the aeronautical features of the past year without offending the Censor and without repeating uselessly what has already been chronicled is a task which, to accomplish satisfactorily and with acceptance to our readers, we feel is beyond our powers. Never before in the history of aeronautics has a period of twelve months been passed through in which more has been done both as regards development of design and the actual use of aircraft than in the year now closed. The first few months of the war taught all the belligerent nations many lessons regarding the use and construction of aircraft, and during the winter months when flying was restricted considerably by the weather conditions all sought to benefit by these lessons. In the spring the result was apparent. Each belligerent may be said to have started the war with aeronautical ideas peculiar to itself, but before the anniversary of the declaration of hostilities had come round a general leveling up has been effected. Each copied the best in the other's practice, and all improved on their own.

The Germans pinned their aeronautical faith to Zeppelins, "drachen" ballons, and aeroplanes of the Taube pattern. We had our biplanes of the B. E. type, and as became a maritime nation, were fast acquiring a predominant position in the matter of seaplanes. France had a few dirigible balloons and a very large number of aeroplanes of many types. Russia had her Sikorski machines—big and little—and was rapidly building up an important fleet of aeroplanes of well-known foreign and British designs. Today we find that Germany has discarded the Taube pattern of aeroplane, while France has thrown out a great number of monoplanes by reputed designers. The "drachen," or kite balloon, soon proved its great usefulness, and is now as much an adjunct of the Allies' as of the enemies' armies. Indeed, both France and Great Britain seem to have established their superiority in the provision of this class of aerial machine. The large Sikorski biplanes have done good work, and have been copied with sundry modifications by all the other belligerents. The day of the twin-engined, twin-screw aeroplane with large span and great lifting capacity has already come. The dirigible balloon has undergone a like development. In Germany the Zeppelin and similar craft are still regarded with favor, and we know for certain that important advances have been made in this branch of aeronautical construction. It is a mistake, however, to suppose that the Germans alone have found a use for the dirigible balloon, although it is quite true that they, and they alone, have seen fit to use them for the destruction of civilian property and non-combatant lives. This country in particular has turned its attention to airship construction, and has evolved and put to good use certain small handy craft. Of these the S. S. type may be specially mentioned.

Such is a general outline of the past year's influence on aeronautics. To fill in the details is, however, as we have said, impossible, for the information necessary to do so is either completely lacking or cannot be made public. All we can do then is to indicate in broad terms the various directions along which aircraft design, under the impetus of war, is developing.

It is now clear that the military use of aeroplanes may be classed under four heads, namely, scouting, gun "spotting," fighting hostile aircraft, and bombing selected spots in the enemy's territory. All these duties cannot be performed efficiently by the one type of machine. The scout and the gun "spotter" must have a clear view of the ground beneath and on all sides of them, and should be provided with signaling means. As it is their duty to avoid combats with the enemy's machines of the third class they should essentially have a high speed and the power of maneuver-

ing rapidly. The machines employed for fighting hostile aircraft clearly require the power of getting off the ground and ascending quickly, and should be armed and armored. The bombing machines should possess great weight-lifting capacity, but as they would be accompanied in their expeditions by protecting machines of the third class they need not be armed with a gun. All four classes must have a wide speed range in order that they may alight slowly and safely on the rough ground of their air base, for the bases at the front are rarely comparable with the aerodromes at home.

In the early days of the war none of the belligerents had done more than begin to contemplate this specialization of types. Accordingly practically the one design in each case had to do all duties or leave those which it could not perform severely alone. However we may criticize the aeronautical policy of our Government in pre-war days, it must be set down that the type of machine, the B. E. biplane, evolved at Farnborough, proved itself better adapted to the actual needs of warfare than those favored by the French or the Germans. France found herself aeronautically in an unfortunate position. She had numerous trained aviators and a large supply of machines. The latter, however, included a big percentage of various designs of monoplanes, which, although perfectly satisfactory in peace, were possessed of most serious drawbacks when it came to war.

Monoplanes—let us take the pre-war Bleriot type, for this was one of the forms that had to be discarded—suffer from at least three distinct military disadvantages. First, their structural strength is considerably less than that of a biplane, which, with its box-girder design of supporting surface, is capable of being constructed on quite strong lines. Secondly, the speed variation possible with a monoplane is small unless the doubtfully sound expedient of varying the angle of incidence of the wings is adopted. Thirdly, the epilot and his observer, if one is carried, have to sit with their heads above the plane of the wings and near the central line thereof. The view of much of the ground beneath them is thus obscured. In spite of the fact that for the same engine power a monoplanes except those of this parasol type, our Ally's aviators were told heavily against France's aeronautical equipment in the earlier stages of the war. The third objection was particularly strong, for it seriously militated against the scouting and gun-spotting duties which then were the chief functions of the aeroplane.

The French aeronautical authorities realized the situation, and as a step towards improving it decided to discard all the monoplanes of the ordinary type then in service. For a time they paid considerable attention to monoplanes of the "parasol" type, that is to say, machines like biplanes with the lower pair of wings cut off. In such craft the pilot and observer sit below the plane of the wings, or in some cases with their eyes just in line with this plane, and so obtain an unobstructed view of the ground beneath. This type of machine had first made its appearance just before war broke out, but it was commonly regarded as inclined to lack sufficient stability owing to the low center of gravity which necessarily accompanied the design. Nevertheless, after the French had excluded all monoplanes except those of this parasol type, our Ally's aviators were able to give a very good account of themselves with the machines at this disposal. It is interesting to note that the machine flown by the late Flight-Lieutenant Warneford when, single-handed, he attacked and destroyed a Zeppelin over Ghent, was a Morane-Saulnier monoplane of the parasol type.

But even this type of monoplane was soon discarded for active service, partly because, like all monoplanes, it was constructionally weak, and partly because of its own peculiar aerodynamic faults, which frequently resulted in side slips. A decree of the French Director of Military

Aeronautics was issued about August, in which it was laid down that biplanes alone should in future be used at the front. Thus France, after a year of war, came to a decision which in this country was reached in 1912, a decision from which our practice has since shown scarcely any departure.

Germany, from the German's point of view, found herself in a still more unsatisfactory position. When war broke out Germany is said to have possessed 700 to 800 monoplanes and 100 biplanes. For two years previously the design of her military monoplanes had been standardized in the shape of the well-known "Tauben." Further, all the later machines were of steel construction, and were fitted with a heavy type of water-cooled engine. Thus when, like the French, they realized the disadvantages of the monoplane in actual warfare—disadvantages which, in their case, were added to by the relative slowness of their heavy machines—they found themselves tied up very considerably by their previous policy of standardization. The accumulated spare parts, the patterns, and the drawings had to be thrown overboard and the construction of biplanes undertaken.

Great Britain started the war with a very excellent type of light fast biplane, and has had whatever satisfaction there be in seeing, not only her Allies, but her enemies hastening to follow her ideas. Nor has it been necessary, or perhaps we should say has it been found possible, greatly to improve on the original pattern. This pattern has been supplemented with machines of two other distinct types, but the B.E.-2 c. machines of today are not materially different from the B.E.-2 machines of August, 1914.

The first important phase of the war's influence on aeroplane design may therefore be said to have been a scramble to discard monoplanes of all types from the flimsiest Bleriot to the heaviest Taube and to adopt tractor screw biplanes of a distinctly British type. France, with her national characteristics, speedily set matters right, and while the Germans were still standarizing their new biplanes, came to the help of this country in maintaining that aerial ascendency over the enemy which we established in the early days of the war. But when all the belligerents were supplied with machines of closely corresponding abilities a new race for supremacy set in. All recognized that, in addition to biplanes of the B.E.-2 type, at least two other forms were desirable. On the one hand there was wanted a small, very fast, easily-controlled type of craft for observation purposes. On the other, there was wanted a machine capable of lifting large weights and flying long distances for bombardment purposes. With these two additional types in existence the B.E.-2 type would be set free to fulfil the intermediate function of fighting the enemy's aircraft.

It has been repeatedly asserted and implied in Parliament and the Press that this country has been backward in following the aeronautical developments of the war, and has in particular allowed Germany to steal a march upon it in this respect. This opinion seems to have been based upon the fact that the first appearance of a large twin-engined, twin-propeller, double-fuselage German biplane was publicly announced towards the end of June in a dispatch from "Eye-Witness," in such a way as to convey the impression that its advent had taken us by surprise. The size and design of the machine were seized upon by certain ignorant persons in order to read the Government a "lesson." Mere size with some people is everything. In this case they even overlooked the fact that the big German machine was driven off by the British machine, presumably a B.E. biplane, and did little more than notice the wonderful descent made by our aviators with their aeroplane on fire after the combat. The real lesson pointed all against size when it came to an

aerial combat, and emphasized once more the excellence of the design of aeroplane with which this country started the war.

The big double-engined machine, with its great weight-lifting capacity, was foreseen and tried some considerable time before the war—for instance, in our issue of July 31st, 1914, a dual-engine arrangement for use with Salmson motors was illustrated. The war demonstrated its military usefulness, and not Germany alone, but all the belligerents, set about developing it. That this country has not been grievously behindhand is indicated by the view of a dual-engined British-built Caudron biplane which we are enabled to give today in our Supplement.

Such machines, however, are, as pointed out above, but one of the two developments which the war has shown to be desirable. The other is the small, fast observation machine for scouting and gun-spotting work. In this direction we were from the first very favorably placed, thanks to the pre-war initiative of our aeroplane manufacturers. The so-called "tabloid" biplane, such as those of the Bristol, Vickers, or Sopwith designs—see "The Engineer" for March 27th, 1914—was in its origin peculiarly British, and it is this type of machine which is now fulfilling so successfully the observation duties at the front. Needless to say, it has been copied by both our Allies and our enemies.

The single-seater "tabloid" biplane is distinguished by its small size, its speed and its large range of speed. A typical British machine of this class has a wing span of 22 feet and a wing chord of 4 feet 3 inches. These small dimensions and the fact that the machine is a biplane permit of its being designed on very strong lines. Its small size, too, makes it very readily handled in the air. For its size it carries a very powerful motor, one of 80 horsepower being fitted in the case we have in mind. The useful load carried is not great—some 340 pounds or, say, the weight of a pilot and fuel and oil for three hours' continuous flight. Its maximum speed is over 95 miles an hour and its speed range is from this figure down to 45 miles an hour. Its climbing speed is very high, being 1,300 feet per minute. We have reason to believe that certain of these figures have recently been improved upon in other British "tabloid" machines, but they are sufficient to indicate the characteristics of the class.

Examining the wings of such a machine we find them given a considerable amount of "stagger." The lower one is set well to the rear of the upper one, and this, with the short wing-chord dimension, gives the pilot a very clear view of the ground beneath him. The upper plane, it will be found, has a much greater angle of incidence than the lower one. This fact, together with the high power of the engine fitted, accounts for the wide speed range and good climbing qualities of the machine. Thus, when the machine is resting on the ground, the upper wing lies at a great angle of attack relatively to the horizontal. Initially, therefore, a great lifting force at a slow speed is developed. To obtain the maximum speed in flight the pilot raises the tail of the machine so that the angle of attack of the upper wing is reduced to a degree or two, while that of the lower wing may be zero or even negative. In either case the lift obtained from the lower wing is, of course, still appreciable. When speed has to be reduced at landing the pilot drops the tail of the machine until the head resistance developed is sufficient.

It is thus clear that the "tabloid" type of machine is very specially fitted to be a scout or gun-spotter, a machine, that is, whose first duty is to see the enemy and not to fight the enemy's machines nor drop bombs on his troops or works.

These three types of aeroplane, the scouting "tabloid," the fighting biplane of the B.E.-2 type, and the bombarding dual-engined machine, thus represent the three points round which the war has caused the aeronautical activity of all the belligerents to crystallize. No doubt other developments

are well in hand. Rumor has it, for instance, that more than one of the warring nations is experimenting with some very large machines carrying more than two engines and crews of half a dozen or so. But it is not yet time to discuss the design or value of such machines.—"The Engineer."

DEVELOPMENT OF THE AEROPLANE.

One of the results of the European war will be a great increase in the size, weight and capacity of the aeroplane and the bringing of it up to the condition where it may be seriously considered for commercial use. The Curtiss Aeroplane Co., Buffalo, N. Y., which built the *America*, is building hydro-aeroplanes of the "Canada" type, having a span of 133 feet, and three planes, each 10 feet wide, and spaced 10 feet apart. It weighs, fully equipped, over 21,000 pounds, and is driven by three twin-six internal-combustion engines of 320 horsepower each. There are three propellers, two in front and one in the rear, each having a length of about 15 feet. In addition, this gigantic air craft will have an auxiliary engine of 40 horsepower, which will be used as a starting engine and for driving a screw propeller for propelling the craft when in the water. It is six times larger than any yet tried, and has a cruising radius of nearly 700 miles at a speed of 75 miles an hour. The weight of the hull is 8,000 pounds and that of the motors 4,000 pounds. It will carry a crew of eight men. Albert Santos Dumont, the famous Brazilian aviator, predicts that in the course of a very few years, giant aeroplanes will be built capable of traveling at the rate of 175 to 250 miles an hour and that regular aeroplane service will be established between the United States and South America. The time required now for the trip from New York to Buenos Ayres is about twenty days, but with the high-speed aeroplane it will require only about two days.—"Machinery."

TOOL-STEEL RESEARCH REPORT.

By Dempster Smith.

(Presented to the Manchester Association of Engineers.)

It will be remembered that the Manchester Association of Engineers, in 1914, appointed a committee to undertake tests in the Heat Treatment of Tool Steel in order to ascertain whether uniformity in results could be obtained in successive trials carried out under identical conditions. The report presented by Mr. Dempster Smith includes opinions by several experts and deals with the results of heat treatment in various stages of hardening and tempering.

To assist in attaining the desired uniformity it was decided that, at the outset, only one grade of tool steel be used. The tools selected had a nose radius of ½ inch; a cutting angle of 70 degrees in a plane at 45 degrees to the length of the tool (this is equivalent to a front and side top rake of 14½ degrees and a clearance angle of 6 degrees. The tools were 1¼ inches square in section, whilst the cut and traverse were to be three-sixteenths inch and one-sixteenth inch respectively, and the cutting speed

to be such as would give a 20 minutes' life to the tool when heat-treated according to the steel makers' instructions. The heat treatment recommended by the maker was to raise the temperature slowly to about 800 degrees C. and then quickly to 1,275 degrees C. From observations made at the steel works it was found that, starting from a cold state, a 1¼-inch square tool took about seven minutes for the first stage and two minutes for the second stage of the treatment. The latter time was deemed sufficient to allow the heat to penetrate to the center of the nose and to fuse the extreme cutting edge. As these tools were heated in the smith's fire, actual observations of the temperatures could not be made.

A two-stage high-speed gas-fired furnace was used for heat treating the tools.

Four tools (Nos. 1, 2, 4 and 5) were forged and ground to approved shape, were heated for 7½ minutes in the low-temperature chamber at 820 degrees C., immediately transferred to the high-temperature chamber at 1,275 degrees C. where they remained for about 1¼ minutes (just long enough to produce a bright glowing strip about 1/16 inch broad along the cutting edge) and afterwards quenched in a molten metallic salt bath at 250 degrees C. The temperatures of the furnace chambers were observed by means of a Féry radiation pyrometer and that of the salt bath by means of an armored mercury thermometer.

For the cutting tests the trials were made on a medium steel bar and without a lubricant. In all cases the test was continued until the tool failed, the nominal depth of cut being 3/16 deep by 1/16 inch traverse. The results showed a slight improvement on re-hardening tool No. 1, and an improvement in one case but not in another for tool No. 5. Tools Nos. 2 and 4 showed a decided deterioration on regrinding only after failure. This result was contrary to expectations, it being generally understood that the performance of a tool improves after the first failure.

An idea is held that high-speed steel is injured by repeated heating to a high temperature such as that required for hardening, and for this reason some engineers prefer to harden turning and similar tools in the smith's hearth, where the heat is concentrated at the nose of the tool. The results of these trials showed that two re-heatings do not appreciably impair the cutting quality of the steel.

It was then decided that tools Nos. 2 and 4 should be normalized, re-ground, hardened and tested against in order to see if the results first obtained by these tools could be equalled or perhaps improved upon. The normalizing (taking about ten minutes) to a dull cherry red (800 degrees C.), placing the tool in hot cinders and allowing it to cool. The normalizing preceded the grinding and hardening processes. In both cases the speed, cut and other conditions were to be the same as prevailed in the first trials with these tools. In neither case did the results equal those obtained by these tools when new, and when the performances of tools which had been simply re-ground and re-hardened after failure were compared with tools which had undergone these processes, in addition to being normalized, it could not be found that any benefit had accrued from normalizing.

It was concluded that in the case of formed cutters which have been unequally strained in machining or in annealing, the normalizing process might tend to eliminate these strains, or perhaps reduce the risk of surface cracks, but it did not seem feasible that this process would have any pronounced effect on a lathe tool which has not been cranked, and which has been carefully forged and not hammered after the temperature has fallen below a cherry red—that is, about 900 degrees C.

The following conclusions were drawn from tests made up to this time:

(*a*) That with freshly-forged tools, hardened as stated and tested under

identical conditions, it was possible to obtain results within 5 per cent. of the mean or within an extreme difference of 9 per cent.

(*b*) That the cutting capacity of a tool which has been tested as stated was not appreciably impaired by being re-hardened as often as three times, but care must be taken to ensure that the whole of the previously injured part is removed in the grinding before re-hardening. In re-hardening a hardened tool the time taken for the cutting edge to reach the glowing stage appeared to be slightly greater than that for a soft tool.

(*c*) That with such tools, treated as stated, normalizing did not appear to have any appreciable effect on the cutting capacity.

(*d*) That the subsequent performance of a tool which had been simply re-ground (after being hardened and failed, as stated) was not equal to the original.

Having regard to the fact that better results are often obtained with a tool which has been simply re-ground after failure than was obtained in the first instance, and that in no case did this occur in these trials, it was decided to obtain the opinion of several steel experts with respect to this phenomenon. One authority who does much testing for the different steel makers could find no fault with the method of heat-treating and testing, or suggest any alteration. He produced the results of a large number of tests of tools, all of which gave better results on being re-ground after failure. A second expert said he obtained the best results from tools heated in an ordinary smith's hearth, and the best final temperature was, in his opinion, between 1,300 degrees C. and 1,350 degrees C. A third authority could find no fault with the procedure or explain why the tool on being re-ground should give better results than in the first trial. He was of opinion that if results were obtained within 10 per cent. of each other the experimenters would do exceedingly well, even with tools made from the same ingot. The manner in which the ingot was cast, heated, and drawn out, also the subsequent annealing process, precluded the possibility of consistent uniformity in the cutting capabilities of the tools. Subsequently, he communicated the following:

(1) It is inadvisable to anneal an ordinary lathe tool after forging and rough grinding. So far as the quality of the tool is concerned, and apart from any possibility of an increased number of wasters, it is preferable to have an ordinary lathe tool made from an unannealed bar.

(2) The pre-heating temperature should be just short of the calescence change point, which in ordinary high-speed steel is rather below 850 degrees C. At such a temperature the time of heating is of no importance except in so far as the surface of the tool may become decarburized. If the pre-heating temperature is above the calescence change point, then in addition to decarburization it is possible also for the crystalline structure to grow larger; but an excess temperature of 20 degrees or 30 degrees C. appears to make no observable difference.

(3) After the pre-heating operation the temperature of the tool should be brought quickly to the final hardening-temperature, say 1,250 degrees C. The reason the heating must be done quickly is because at these high temperatures the crystalline structure grows coarser at a very quick rate, and in five minutes or less many kinds of tools would be entirely spoilt.

(4) Certain kinds of high-speed steel air-harden intensively, and tools of moderate dimensions, or toothed tools made from such steels, had better be air-hardened. But there are other steels which require oil-hardening in order to get anything like the best results. There are also steels which if oil-quenched are soft, but if air-cooled are much harder.

A fourth authority stated: (1) That he got the best results by heating to near the fusing point and cooling in air, but this was only practicable on tools that could be easily ground up if the point was injured; this method of heating was safe and should result in uniformity as far as

the hardening was concerned. (2) As regards re-grinding after failure, this obviously depended on the cause of failure. If the tool, as often happened, was so hard that the edge "crumbled" instead of cutting, re-grinding usually gave better results, but if the tool was not hard enough re-grinding was of no use.

A fifth correspondent wrote that: (1) The possibility of the nose of the tools getting slightly hotter than the other parts, thereby fusing the extreme edge, required that extra grinding should be done on such a tool. (2) It was a well-known fact that when a salt bath had been in use for some time, it became contaminated with particles of iron oxide, which floated about and might possibly attack any tools placed in the bath. (3) With regard to the advisability of re-hardening after the tool had once lost its cutting edge, it was his experience that this was not necessary, as in a great many instances it was found that a high-speed tool cut better and did far more work after the first regrinding.

The sixth authority consulted was of the opinion that the tools had been handled too delicately, and that for good results in subsequent re-grindings the tools should be heated until the surface was blistered for a distance of about 3/8 inch from the nose. Of all the suggestions this seemed the most reasonable, and it was decided to take two new tools (Nos. 3 and 6), also two old tools (Nos. 2 and 4), treat as recommended, and test as before. After failure one old (No. 2) and one new tool (No. 6) were to be re-ground, whilst tools 3 and 4 were to be re-ground and re-hardened and all tested again. The tools were heated for ten minutes (starting from cold) in the pre-heat chamber at 825 degrees C., and from 2¼ to 2½ minutes in the final-heat chamber at 1,275 degrees C. The latter time was required in order to produce the desired degree of blistering, and is about half a minute in excess of the time previously given to the tools at this stage. The tools so treated on being chipped at the cutting edge showed a somewhat coarse structure, the crystals being fully twice the size of those obtained in tools when treated as in the first instance.

The trials after treatment were more irregular and, on the whole, inferior to the results obtained in the early trials with the old method of heat treatment. Considering that consistent uniformity was the object aimed at, the results in this series were disappointing.

With the object of obtaining uniform results it was resolved to revert to the old limits of heat treating, and to cool the tools in an ordinary air blast, instead of the molten salts 250 degrees C., which had been used in all the tests up to this stage; but the results of this series showed no improvement on the first trials, and proved that uniformity was not to be obtained by normalizing, and that however carefully the tools might be treated irregularities appeared to be unavoidable.

The remainder of the report records the Committee's experiences in watching expert hardeners at work and the results of sundry cutting tests.

The Committee's conclusions are as follows:

(1) Normalizing shows no advantage for turning tools.

(2) A tool which has only been raised to the hardening temperature at the cutting edge will deteriorate on re-grinding, whilst one that has been overheated at the cutting edge will improve on re-grinding.

(3) The methods of hardening adopted by experts show great variation both as to furnace temperature and duration of treatment, but this report will give those interested some definite guidance in both these particulars.—"Page's Engineering Weekly."

FLOW OF STEAM IN PIPES.

By W. L. Durand.

SYNOPSIS.—Method of determining the sizes of steam pipes by the use of formulas and by a curved chart by which, if three of the four variables are known, the fourth can be found.

Generally, boilers, engines and accessories are chosen with care, but when it comes to such items as piping, this is either guessed at or designed on the velocity basis, without regard to the factors entering into the flow of steam. Yet what is more important in the design of engineering work than the piping? The commonest method for determining the size of steam pipes is to assume a velocity of 6,000 feet per minute for high-pressure pipe and 4,000 feet for low-pressure pipe. That this method will not give satisfactory results, except over a very limited range, may easily be seen if the drop in pressure at these velocities is computed for different sizes of pipe. The accompanying table is based on a length of pipe of 100 feet and an initial pressure of 100 pounds at 6,000 feet per minute.

Size of pipe, inches.....	1	1¼	1½	2	3	4	6	8	10
Loss of pressure, pounds.....	16	10	7.7	5	2.6	1.7	1.0	0.68	0.5

From the foregoing it may be seen that in small pipes the loss in friction is so great as to affect the economy of operation; on the other hand, in the large sizes of steam lines bigger pipes are specified than are necessary. This increases the cost of installation to an appreciable extent since the cost of pipe and fittings increases rapidly with the size.

The factors that enter into any problem of this character are weight or quantity of steam, length of pipe, density of the steam, drop in pressure and the diameter of the pipe. To obtain accurate results it is necessary to take into consideration all of the foregoing factors, and any formula used in this connection should contain all of them.

The most commonly accepted formula for the flow of steam is Unwin's, which is as follows:

$$W = 87.5 \sqrt{\frac{Dpd^5}{L\left(1+\frac{3.6}{d}\right)}}$$

where

W = Weight of steam flowing in pounds per minute;
L = Length of pipe in feet;
d = Internal diameter of pipe;
D = Density, pounds per cubic foot;
p = Drop in pressure, pounds per square inch.

The reason that this formula is not more generally used is presumably because of the excessive time required in solving it. Recently the writer published a table based on dividing Unwin's formula into three parts. The length of pipe and friction loss may be reduced to one variable "friction loss per unit length," such as "per hundred feet."

$$W = 87.5 \sqrt{\frac{p}{100}} \times \sqrt{D} \times \sqrt{\frac{d^5}{1+\frac{3.6}{d}}}$$

This table is reproduced here to show its application :

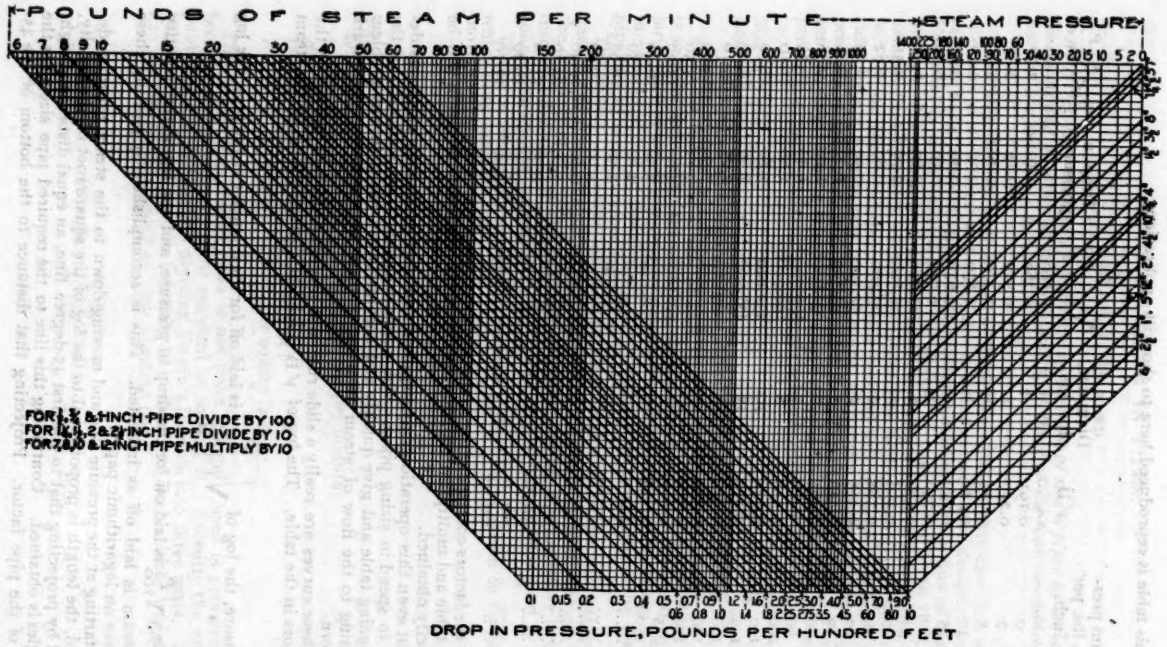
Steam pressure, lbs. per sq. inch.	$\sqrt{D}$	Diameter, pipe, ins.	$\sqrt{\frac{d^5}{1+\frac{3.6}{d}}}$	Drop in pressure.	$87.5\sqrt{\frac{P}{100}}$
.....	.....	¼	0.116	.....	.....
0	0.193	½	0.262	0.10	2.77
2	0.207	1	0.522	0.15	3.39
5	0.223	1¼	1.170	0.20	3.91
10	0.248	1½	1.830	0.30	4.79
15	0.270	2	3.670	0.40	5.53
20	0.290	2½	6.050	0.50	6 19
30	0.326	3	11.100	0.60	6.77
40	0.358	3½	16.800	0.70	7.31
50	0.388	4	23.500	0.80	7.82
60	0.415	4½	32.100	0 90	8.29
70	0.440	5	43.500	1.00	8.75
80	0.464	6	71,600	1.20	9.58
90	0.486	7	106.000	1.40	10 35
100	0.508	8	150.000	1.60	11.10
120	0.548	10	272.000	1.80	11.70
140	0.585	12	437.000	2.00	12.40
160	0.620	...	.....	2.25	13.10
180	0.653	...	.....	2.50	13.80
200	0.684	...	.....	2.75	14.50
225	0.721	...	....	3.00	15.20
250	0.756	...	.....	3.50	16.40
.....	.....	...	.....	4.00	17.50
.....	.....	...	....	4.50	18.60
.....	.....	...	.....	5.00	19.60
.....	.....	...	.....	6.00	21.40
.....	.....	...	.....	7.00	23.10
.....	.....	...	.....	8.00	24.70
.....	.....	...	.. ..	9.00	26.20
.....	.....	...	.....	10.00	27.70

If the factors corresponding to the variables in any problem are found from this table and multiplied together, the number of pounds of steam will be directly obtained.

But even this operation may require more time than many engineers desire to spend in sizing pipes. The accompanying curves are based on the foregoing table and give quickly and accurately the result in any problem relating to the flow of steam in pipes if three of the four variables are known.

These curves are really a slide rule and mechanically add the log of the factors in the table. The log of $\sqrt{D}$ is laid off corresponding to the steam pressure, the log of $\sqrt{\frac{d^5}{1+\frac{3.6}{d}}}$ is laid off for the diameter of the pipe, the log of $87.5\sqrt{\frac{P}{100}}$ is laid off for the drop in pressure, and the log of the quantity of steam is laid off as indicated. This is accomplished by plotting these curves on logarithmic paper.

Starting at the pressure line and moving down to the steam pressure desired, the length is proportional to the log of the square root of the density, and by projecting that to the first 45-degree line, an equal distance at right angles is obtained. Continuing this line to the required pipe size adds the log of the pipe factor. Projecting that distance to the bottom of the 45-


POUNDS OF STEAM PER MINUTE
STEAM PRESSURE
FOR ¼, ⅜ & ½INCH PIPE DIVIDE BY 100
FOR ¾, 1, 1¼, 2 & 2½INCH PIPE DIVIDE BY 10
FOR 3, 3½ & 4INCH PIPE MULTIPLY BY 10
DROP IN PRESSURE, POUNDS PER HUNDRED FEET

CURVES BY WHICH IF ANY THREE VARIABLES ARE KNOWN THE FOURTH CAN BE FOUND.

degree lines representing the drop in pressure, a distance equal to the sum of the pressure and diameter is obtained on the vertical scale. By moving up to the required drop the log of the factor for the pressure drop is added, and projecting the sum of the logs of the three factors, the answer is read directly on the quantity scale in pounds of steam per minute.

In order to show the use of this chart, take a hypothetical case of a steam main 800 feet long and 4 inches diameter with steam at 100 pounds pressure. What will be the flow of steam if the drop is limited to 4 pounds? The drop per 100 feet is ½ pound. Starting at 100 pounds and running down to the 4-inch diagonal line and over to ½ pound, 73 pounds of steam per minute is read on the "pounds of steam" scale. Suppose the problem to be the same except the pounds of steam given and the drop was to be found. Starting at 100 pounds and running down to the 4-inch diagonal line and over to 73 pounds, ½ pound is found at the intersection. Given any three of the four variables, the fourth can be found from this chart.—"Power."

THE SPERRY PALLOGRAPH.

CONSTRUCTED BY THE SPERRY GYROSCOPE COMPANY, BROOKLYN, N. Y.

The only pallograph in America, and, indeed, one of but two or three in existence, has lately been completed by the Sperry Gyroscope Company, Brooklyn, N. Y., for the use of the United States Navy engineers at the Model Basin, Washington, D. C., in connection with their investigation of vibrations of ships. Briefly, the pallograph is an instrument which simultaneously records vertical and horizontal transverse vibrations; and while the instrument is primarily intended for use aboard ships, it may be used to trace vibrations or oscillations of any character to their primal source. Doubtless the first practical pallograph was the one made some years ago by Dr. Schlick, of Hamburg, and it was he who demonstrated the wonderful results that may be achieved by means of this device in definitely locating the source or cause of vibrations.

A pallograph consists of six principal parts, namely:

(*a*) Apparatus for measuring vertical vibrations.
(*b*) Apparatus for measuring horizontal vibrations.
(*c*) Mechanism for propelling the recording strip.
(*d*) Electrical connection with ship's tail shafts.
(*e*) Clockwork for registering seconds; and
(*f*) Marking pens.

When the pallograph is set up on a vessel where it is moved up and down by vertical vibrations, a weight, shown diagrammatically at W, in Fig. 1, by reason of its inertia remains at rest, although it appears to

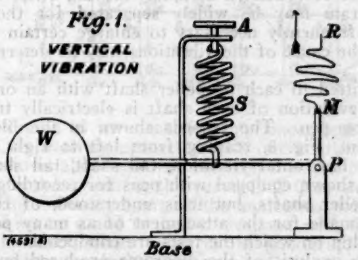
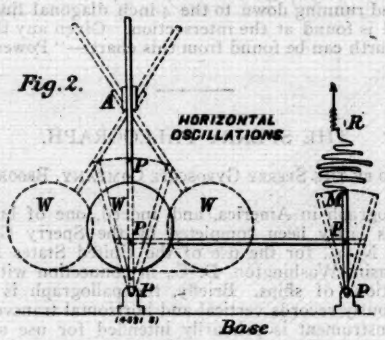


Fig. 1.
VERTICAL VIBRATION

move rapidly up and down. When the apparatus is in a state of rest, a vertical impulse given to the weight will cause it to swing up and down with oscillations having a certain regular period. When measurements of vibrations are to be made, this period must always differ materially from the period of the vibrations which are to be analyzed; otherwise the weight would begin a synchronous oscillation, and anything like a record of vibration would be impossible. The period of the weight is quickly changed, however, by varying the tension of the spring S and moving it to or from the point marked P.

The pendulum used to record horizontal oscillations is diagrammatically illustrated in Fig. 2. It will be noted that the swivelled sleeve A is



adjustable in a restrained vertical direction, and it will be readily understood how, by raising or lowering this sleeve, the pendulum may be lengthened or shortened—every slight adjustment of the sleeve is tremendously multiplied in its effect near the critical point—on the one hand representing a pendulum hung from a height of several hundred feet, and on the other almost zero height. Indeed, in the Sperry pallograph it is possible to arrange the parts so that the weight acts as if suspended from a practcially infinite height, thus creating a pendulum of many miles in length, and the length of this pendulum may be adjusted from zero to maximum in two or three seconds.

Records are made on a 5-inch paper strip, which is moved upwardly at constant speed by a small motor; the speed of travel may be regulated from 1½ inches down to ⅝ inch per second, as required. It will be readily understood that vibrations of high frequency or of considerable amplitude should be recorded on a rapidly-moving strip, so that the wavy lines of the diagram may be widely separated for thorough analysis; indeed, it is even frequently necessary to enlarge certain sections, so that conclusions as to the cause of the vibrations may be determined with absolute certainty.

A collar being fitted to each propeller shaft with an ordinary electrical connection, each revolution of each shaft is electrically transmitted to its respective recording pen. The records shown in the illustration of the complete instrument, Fig. 3, reading from left to right, are as follow: Vertical vibration, horizontal vibration, tail shaft, tail shaft and seconds. The instrument is shown equipped with pens for recording the revolutions of only two propeller shafts, but it is understood, of course, that provisions have been made for the attachment of as many pens as there are tail shafts in the ship on which the tests are conducted.

In discussing the analysis of the diagrams produced by the pallograph,

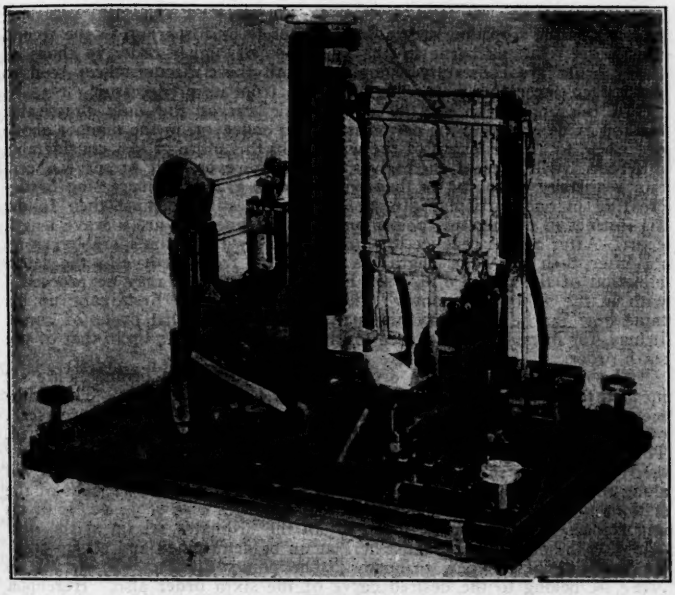
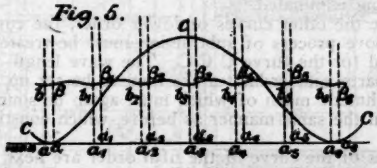


FIG. 3.

Dr. Schlick stated: "It is necessary, in the first place, to exactly determine the period within which the ordinates always have the same value. When the curve is a regular one the determination of the period is a simple matter. If, for instance, the curve C C C in Fig. 4 be given, tangents

Fig. 4

Fig. 5.



a a and *b b* are drawn to its successive uppermost and lowermost points, so that these are respectively connected with one another. Then a line *c c* is drawn in a position about midway between *a a* and *b b*, as may appear suitable, parallel to these, cutting the curve C C C in the points *m*, *n*, m_1, n_1, m_2, n_2, &c. The distances apart of the analogous points of intersection *m* and m_1, m_1 and m_2, or *n* and n_1 then correspond with the period.

"The curves, however, which occur in the pallographic investigation do not show the complete regularity here assumed, at any rate in the case of

HALF SECTION A.B. LOOKING FORWARD. END ELEVATION LOOKING FORWARD.

Fig. 2.

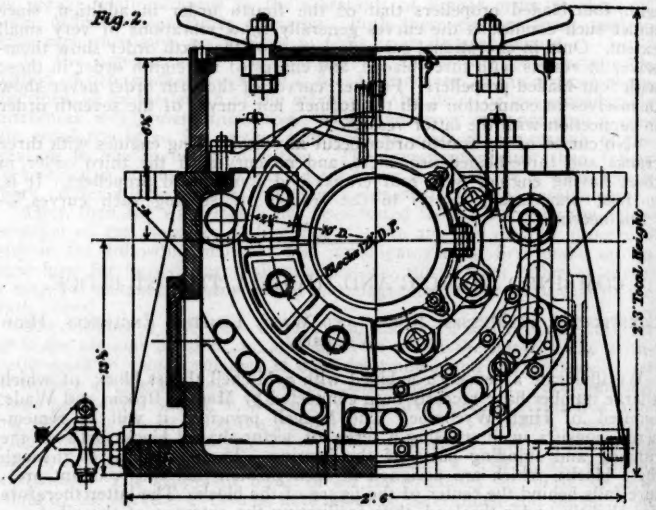


Fig. 3.

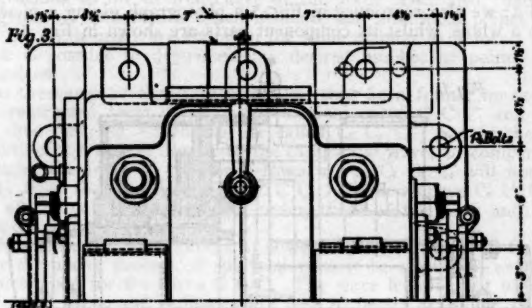
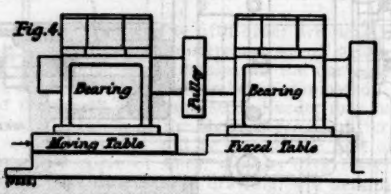


Fig. 4.



It will be seen from Fig. 1 that the shaft rests on two white-metal bearings, between which is the single collar on which the whole of the thrust is taken. The Michell blocks, as shown in Fig. 2, extend round a portion of the shaft only, the arrangement being thus in some sense equivalent to the common horseshoe type of thrust-block collar, and affording the same facilities for the examination of the bearing when desired. Each Michell block is supported by a screw stud, as best seen in Fig. 1. When erected these studs are tightened up until each takes an equal bearing, and they are then locked by means of the nuts shown. As will be seen from Fig. 2, the housing is carried well up above the center line of the shaft, which makes it possible to get in a couple of blocks above this level, so that the "horseshoe" extends round more than half the total circumference.

To prevent the blocks twisting round their points of support they are enclosed in a cage, as indicated in Fig. 2, from which, moreover, it will

FIG. 5.

be seen that the cage is itself made in halves. The upper half can, with the two blocks it confines, be lifted out bodily when the set-bolts taking the thrust are run back, and the lower half can then be pushed round and taken out in its turn. The two cages can be seen amongst the other components in Fig. 6, where one is shown with the blocks in place, and the other with them omitted. With the engine running ahead, the thrust is taken by the blocks on the forward side of the thrust collar, a second set on the after side taking the thrust when the engine is running astern. An oil well is formed at the bottom of the housing, and the collar, dipping into this oil, picks it up and distributes it over the whole of the surfaces to be lubricated. A stuffing box at the outer end of each bearing prevents loss of oil by leakage, while the top of the housing is closed by an oiltight cover, as indicated.

The thrust block in question has been built for a shaft making about 400 revolutions per minute; and to carry off the heat generated in consequence of this high speed, cooling coils, through which water is pumped, are arranged in the oil well, as indicated.

The shaft has a diameter of 7½ inches, and the collar one of 15 inches, whilst the area of one set of blocks is 50 square inches.

A four-hours' test of the bearing was made by Messrs. Broom and Wade at speeds and loads considerably in excess of those which it is intended to take in service. The method adopted was to mount two blocks in alignment with each other, and to push the two together by hydraulic jacks. The arrangement is shown diagrammatically in Fig. 4. The shaft was driven by belting from an electric motor, which, running light, took 2

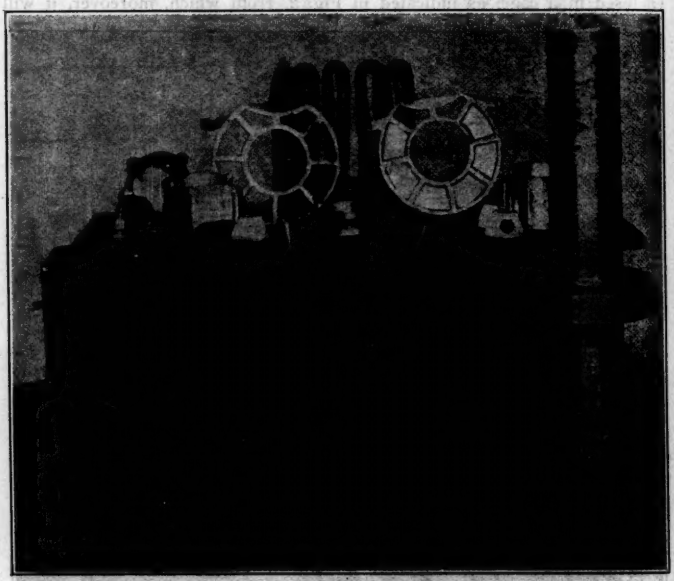


FIG. 6.

ampères of current. Coupled up to the shaft, but with no thrust on the blocks, the current consumed was 8 ampères. The observations made are recorded in the table annexed.

The temperature of the collar at the end of the test was 112 degrees F. The cooling water was circulated at the rate of 6.5 pounds per minute. At the outset the bearing pressure was 300 pounds per square inch, which was increased to 400 pounds per square inch after 2½ hours' run.

From the power absorbed by the motor at 400 revolutions per minute the coefficient of friction can be calculated and a value of 0.0015 is obtained, whilst from the temperature rise of the cooling water the value 0.0012 is deduced. The total friction loss on the two bearings as thus measured amounted to 214 B.t.u. per minute. Making allowance for some loss of heat by radiation, the two estimates are, it will be seen, in excellent agree-

ment. The ordinary type of thrust-block has a coefficient of friction fully twenty-five times as great.

FOUR HOURS' TEST OF CENTER NO. 4 MICHELL THRUST BEARING, CARRIED OUT BY MESSRS. BROOM AND WADE, LIMITED, HIGH WYCOMBE, ON AUGUST 19, 1915.

Time.	Amperes.	Volts.	Revolutions per minute.	Total Load, in pounds.	Oil temperature. Center No. 4.	Water temperature.		Journal bearing temperature. Center No. 4.	
						In.	Out.	Forward.	Aft.
					deg.	deg.	deg.	deg.	deg.
6.15	17	420	425	15,000	63	57	70	87	84
6.30	17	420	435	15,000	...	57	73	89	86
6.45	17	420	435	15,000	...	57	76	92	90
7.0	17	420	440	15,000	...	57	82	95	92
7.15	17	420	440	15,000	...	57	84	96	93
7.30	17	420	445	15,000	...	57	85	97	94
7.45	15	420	445	15,000	...	57	85	97	95
8.0	15	420	445	15,000	...	57	86	98	95
8.15	15	420	450	15,000	...	57	86	98	95
8.30	15	450	455	15,000	...	57	86	98	95
8.45	14	450	455	20,000	...	57	86	98	95
9.0	14	450	460	20,000	...	57	86	98	96
9.15	14	450	460	20,000	...	57	88	98	96
9.30	14	450	460	20,000	...	57	88	98	96
9.45	14	450	460	20,000	...	57	89	99	97
10.0	14	450	460	20,000	...	57	90	100	98
10.15	14	450	460	20,000	80	57	90	100	98

We may perhaps add that with such speeds as are usual in the ordinary merchant service cooling of the oil would have been unnecessary, as radiation and convection suffice to keep the bearing cool, provided that the heat generated does not exceed about 50 B.t.u. per minute in a bearing of this size. In the present case each bearing generated heat at the rate of about 107 B.t.u. per minute.—"Engineering."

MANUFACTURING SEAMLESS STEEL TUBES.

By J. J. DUNN (SHELBY STEEL TUBE COMPANY, PITTSBURGH, PA.).

Read before the Canadian Railway Club, Montreal.

As it seems probable that the cost of seamless tubes will gradually approach the cost of welded tubes, and each improvement made will open up a wider field for their use, it will doubtless be of general interest to describe the various processes employed for making seamless tubes of steel. It is as important to understand the advantages and limitations of

the seamless tube as it is of any other engineering material, for indiscriminate use of this material tends to discourage its use in those places where by its character it is fitted to give economical service. The advantages of the seamless tube are almost self-evident, but its limitations are most surely comprehended only when the manfacturing methods employed in its making are clearly understood.

Seamless tubes are made today by two general processes: the piercing process and the cupping process. As the piercing process is used for by far the larger portion of seamless products it will be first described. The cupping process will be but briefly referred to.

There are two general methods of piercing, first rotary piercing, and second, punching. The punching method is used for special classes of work, but it is the rotary piercing process that produces the larger tonnage and is the most widely used. The possibility of rotary piercing was discovered by Mannesman, who devised practically all of the many combinations of rolls that have been successfully used. There is, however, one notable exception which is the arrangement proposed and used by R. C. Stiefel.

ROTARY PIERCING.

The essential features of a machine for rotary piercing are revolving surfaces so arranged that their contact with the cylindrical piece of metal which is to be pierced will impart to it a simultaneous motion of rotation and translation; and a mandrel for controlling the flow of the metal. To accomplish this it is sufficient that the axes of the rolls and of the cylinder of metal be neither parallel nor intersecting lines. Practically the departure from the condition of parallel or intersecting axes must be wide enough to overcome the frictional resistance to motion in translation and not so wide as to prevent rotation.

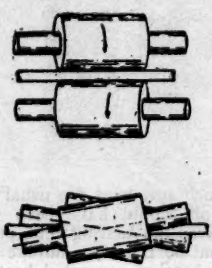


FIG. 1.

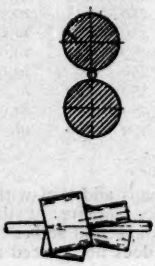


FIG. 2.

Fig. 1 shows in elevation and plan, cylindrical rolls which will impart the desired motion to a cylindrical blank between the rolls and in contact with them. Fig. 2 is an elevation and plan of sections of the rolls by planes at right angles to the axes of the rolls and makes clear how the rolls impart the two motions to the blank between them. These rolls are not a suitable arrangement for piercing as they can have only a small area of contact with the blank.

Fig. 3 shows a practicable arrangement. In this figure the solid blank B is shown between the rolls and partly pierced. The piercing mandrel C is in its working position supported by a bar D. The rolls A are truncated cones so proportioned as to properly distribute the work of reducing the metal of the solid blank as it passes over the piercing mandrel and is formed into a tubular section.

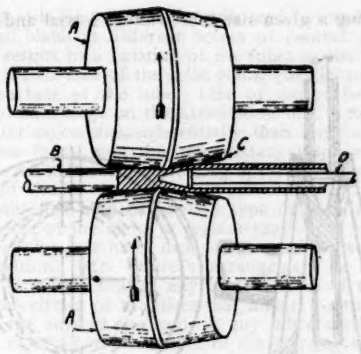


FIG. 3.

ROLL OPERATION.

In operation the rolls are adjusted to the proper distance apart and the mandrel is placed in position. The rolls being in motion, the heated blank is pushed into the entrance angle and, coming in contact with the rolls, takes motion from them of rotation and longitudinal travel. The first work of the rolls is to reduce the diameter of the blank, giving it a conical form and increasing its contact with the rolls as it encounters the piercing mandrel. Then the reduction of cross-sectional area of the blank takes place very rapidly as it passes over the mandrel. The diverging angle of the exit side of the rolls permits equalizing the work along the length of the mandrel and also gives room for the expansion in diameter that is an effect of the rolling contact. Further it permits of such a relation of the surfaces of the rolls and the mandrel as to produce a comparatively smooth pierced billet.

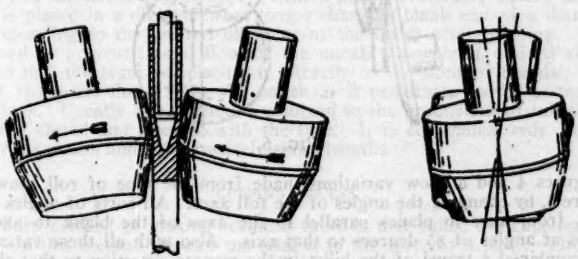


FIG. 4.

It must be evident that the angles of entrance and exit passes of the rolls; the diameter and angles of the mandrel; the relation of the position of the mandrel to the roll passes; the diameter of the blank; and the distance between the rolls influence profoundly the size and character of the tube produced. Besides these factors, the heat of the metal being worked; the speed of the rolls; the relative rate of longitudinal travel to rotation, all have a decided effect. Indeed, it is only within the last ten years that the relations of these various factors were well enough understood to

permit of producing a given size tube without a trial and error adjustment of the machine.

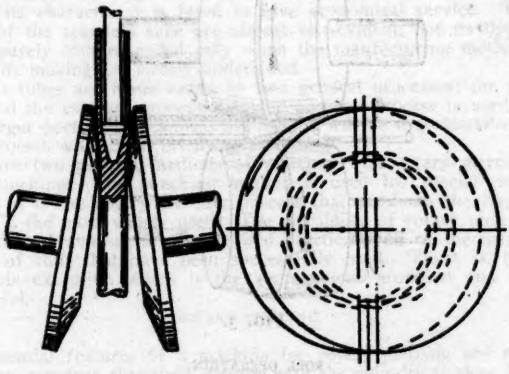


FIG. 5.

The divergence from the desired result occurred in the diameter and wall thickness of the hollow billet produced; in the concentricity of the bore and in unnecessary working of the metal. In the efforts made to minimize these variations many designs of piercing machines were worked out and not a few of them were actually constructed.

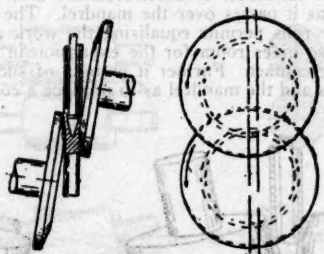


FIG. 6.

Figures 4 and 5 show variations made from the type of roll shown in Figure 3, by changing the angles of the roll axes. All sorts of angles were tried, from axes in planes parallel to the axes of the blank to axes in planes at angles of 85 degrees to that axis. Also with all these variations was combined a travel of the billet in the reverse direction to that shown in the figures together with a wide range of entrance and exit angles, as well as variations in the shape of the piercing mandrel. Practically all these various arrangements will pierce a solid blank, and with each type of machine there is a combination of roll or disc diameter, entrance and exit angles, mandrel angles and feed which will produce good work on at least a limited range of sizes.

MINIMUM DISPLACEMENT OF METAL REQUIREMENT.

One of the most important requirements in a piercing machine is that the tube be formed with a minimum displacement of metal. Looking again

at Figs. 3, 4 and 5, it is evident that the rolls tend to impart different speeds of rotation to the blank at different points of contact along the axis of the blank. This results in a twisting of the tubes as the piercing proceeds, so that a line on the surface of the solid blank, parallel to its axis, becomes a helix on the surface of the tube. One of the earlier patents claimed this feature as an advantage on the assumption that a tube is usually subjected to a greater stress circumferentially than longitudinally, but practically it has been found that this unnecessary displacement of material tended to rupture the metal in the piercing operation should there exist any defect in the solid blank.

R. C. Stiefel designed and patented a type of piercing machine which reduced this twisting of the metal in regular practice to such an extent that the helix was right or left hand depending upon the setting of the discs and mandrel position. Mr. Stiefel's arrangement is shown in Fig. 6. From the figure it is seen that at any cross section of the blank the sum of the radii of the circles of the discs that impart rotation to the blank is exactly equal to the sum of the radii at any other section; hence, except as the motion is modified by the action of the piercing mandrel, the blank, solid or pierced, is revolving at a uniform rate at every section.

It will be noticed that there is a very considerable slip between the billet and the discs, varying from zero at a point midway between the disc axes to a maximum at the points where contact between discs and blank ceases. This slip has no very important influence on the work. However, it causes a waste of power, a disadvantage that is more than offset by the other good features of the design.

These points of the rotary-piercing operation have been dwelt on at some length, as it is this operation alone that is peculiar to the making of seamless tubes, and an understanding of the numerous factors that modify the results is necessary to an understanding of succeeding operations and to resulting variations in the finished product.

THE ERHARDT PUNCHING PROCESS.

For certain classes of work a process of piercing is used that consists of simply punching the hole in the solid blank. This method is generally known as the Erhardt process. A heated blank, preferably square in section, is placed in a die somewhat longer than the blank end of a diameter corresponding to the desired diameter of the blank after piercing. Thus confined to prevent lateral flow of the metal, a punch of desired size is forced into the blank, displacing it laterally to the limit of the die, after which the metal flows along the punch as it penetrates to the bottom of the blank. Usually this method is confined to the production of tubes with one end closed and integral with the tube. It is economical only for the larger diameters and comparatively short lengths.

CUPPING PROCESS.

A third method of making seamless tubes is that called the cupping process. In this method the raw material is in the form of a plate. The first operation is to trim the plate to a circle by which at the very beginning a loss in scrap of 25 per cent. occurs. After trimming, the plate while hot it is pushed through a circular die and formed into a cup approximately hemispherical in section. Successive passes through dies of decreasing diameter elongates the cup into a tube with one end closed. The economical application of the cupping process is for tubes of large diameters and heavy wall. Their only limit seems to be the size of the presses required for the work. Tubes of 24 inches in diameter or even larger, weighing 6,000 pounds each, can be produced on existing equipment.

The blank after piercing varies so greatly in dimensions of both the diameter and wall thickness that additional operations are required to bring

it within the variations required by most uses. For many purposes the required accuracy can be attained by a hot operation, but for others requiring still closer limits resort is had to cold working. Succeeding operations are required for production of sizes which cannot be accomplished by piercing alone.

Many schemes have been advised for the hot finishing of the pierced tube, rolling on long, solid mandrels in various ways, extracting the mandrel after the operation; drawing through dies and over long mandrels, and rolling on short mandrels have all been thoroughly tried out more or less successfully, but it is rolling on a short mandrel that has had the widest application, on account of the accuracy of the product and the comparatively low cost of the operation. Fig. 7 shows diagrammatically a machine for this method.

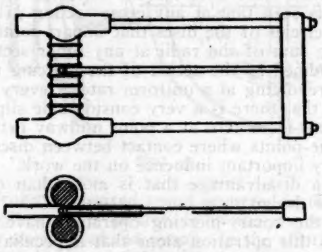


FIG. 7.

A stand of two high rolls with circular grooves is used. A short mandrel is supported in the groove of the rolls by a long mandrel abutted against a yoke which is tied by long rods to the roll housing. In operation the pierced billet at a proper temperature is presented to the rolls, which seize it and force it through the groove and over the mandrel. As the annular area between the mandrel and roll groove is less than the area of the cross section of the pierced billet, a reduction in area and corresponding elongation takes place. After the tube has passed the rolls and is on the long mandrel, the mandrel is pulled forward free from the tube. The tube is then raised and the mandrel is passed back again into the rolling position while the tube passes over the top of the rolls for the next pass. Successive passes are made in this manner until the desired wall thickness and diameter is obtained, reheating the billet as may be required.

This is the old hand method of rolling and is still used to some extent, as it is easy to handle the tube to grooves of successively decreasing diameter using several mandrels. The method is slow and laborious and has been largely superseded by a mill in which all hand handling is eliminated. To accomplish this some slight changes were necessary in the rolling method. These changes consist principally in the use of but a single groove and mandrel. The varying area of the pass is obtained by simply raising rolls for the earlier passes and using a slightly smaller mandrel. The finishing passes are taken with the rolls in a fixed position and with the required sized mandrel for the finished inside diameter. The long mandrel is fixed permanently in position and the short mandrel only is removed after the passes.

To return the tube to the entering side of the rolls for the next pass the top roll is elevated and the tube stripped from the long mandrel by an auxiliary set of rolls. On the entering side of the rolls the tube is received in a V-shaped trough in which it is moved to the rolls by a pusher operated by compressed air. All these operations are mechanical except the removal of the short mandrel.

HOT-FINISHED SEAMLESS BOILER TUBES.

The successful introduction of this mechanically-operated mill made possible the "hot-finished" seamless tube in the boiler-tube sizes. Hand handling was too slow to allow the required thickness of wall to be obtained in one heat, which is an essential requirement on account of finish. Rolling reduces the wall thickness of the pierced billet to gages lighter than can be obtained by the piercing operation and gives the accuracy required for boiler tubes and other similar work. The sliding of the hot tube over the stationary mandrel, however, causes scratches on the interior wall of the tube that are objectionable for some purposes and in all cases spoil the appearance of the finished tube. These scratches are removed by a following operation, which has come to be called "Reeling."

The reeling machine is very similar to a piercing mill of the roll type with axes in parallel planes, such as shown in Fig. 1, but the rolls of the reeling machine are shaped to give contact over a considerable length of tube, usually not less than a length somewhat greater than the tube diameter. The mandrel for reeling is cylindrical instead of conical as for piercing. The action of the rolls in giving the tube rotation and longitudinal travel is the same. Fig. 8 shows the arrangement.

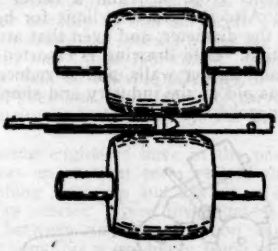


FIG. 8.

After the tube has been brought to required diameter and wall thickness by the rolling mill, it is conveyed to the reeling machine. Here the surfaces, interior and exterior, are rolled in a direction approximately at right angles with the direction of rolling on the two-high mill, resulting in the practical elimination of any scratches caused by the rolling mandrel. At the same time the tube is rounded up and the outside burnished. The reeling machine introduces a variation in the pressure of the reeling rolls and in the heat, and this variation is corrected by passing the tube after reeling through a set of rolls with circular grooves of a diameter the same as the required finished diameter plus an allowance for shrinking in cooling.

RECAPITULATION.

These are the standard hot operations in making a seamless steel tube. The piercing operation forms a tube from the solid blank. This tube varies greatly in diameter and wall, not only from the tube to tube but in the same tube at different points of its length. The rolling operation reduces the rough-pierced tube to accurate circumference and wall thickness, but the rolled tube is not round nor is it smooth on the interior wall. The reeling machine smooths the walls and rounds the tube but introduces a silght variation in the circumference, which is removed by the final sizing operation. Thus it is seen that each operation, after the piercing, is for the purpose of correcting and perfecting the preceding operation. It is not essential that these operations be separated and placed on different

machines in this way, as it is entirely possible to complete the work in the piercing operations, but such a nicety of adjustment would be required to control simultaneously all the different factors that it has been found simpler and cheaper to make a separation as described. There is one variation occurring in the piercing operation that is not removed to any appreciable extent by the succeeding operations, that is, the eccentricity of the bore of the tube with the outside surface. The eccentricity varies directly as the wall thickness and is influenced by a number of factors—the heat of the blank; the ratio of longitudinal feed to rotation; the form of the rolls and mandrel and of the supporting guides all have an effect. Even when all these factors are as constant as it is possible to maintain them, a variation in eccentricity occurs. Under good conditions, however, about 50 per cent. of the product will be not more than 5 per cent. eccentric. The established limit for inspection is 10 per cent. eccentricity. This tendency to eccentricity must be taken into consideration in applying seamless tubes. Fortunately for most uses the maximum variation causes no serious inconvenience.

DRAWING-BENCH FEATURES.

There are numerous uses to which seamless steel tubes are applied that require smaller variations from size and a better surface than can be obtained by hot work. Also the practical limit for hot work is a tube not less than 1½ inches in the diameter, and even that at this time is too small for economical production. Cold drawing is resorted to in order to obtain the smaller diameters and lighter walls, and to reduce variations in dimensions. The process is as old as the industry and simple in details, although requiring skill to obtain good work.

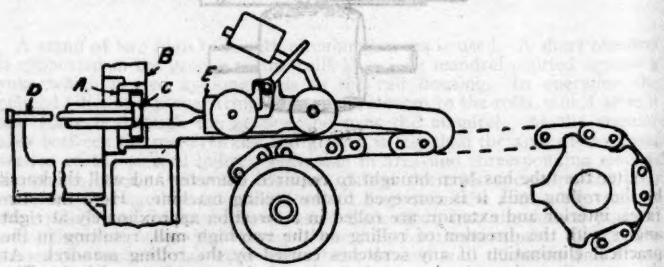


FIG. 9.

Fig. 9 illustrates the essential features of a "drawing bench." A die A of the required size is supported in a die head B. This die controls the outside dimensions of the tube. Within the die is placed a mandrel C which controls the inside dimensions. The mandrel is anchored to the rear end of the bench by the rod D to prevent longitudinal movement. In front of the die is a gripping mechanism E by which the tube is pulled through the die. This gripping mechanism is moved by attaching it to a moving chain or to the piston of a hydraulic cylinder.

For cold drawing the hot-rolled tube is prepared by reducing the diameter of one end by a forging operation. This is for the purpose of allowing the tube to pass through the die far enough to be seized by the gripping mechanism. After thus "pointing," the tube is pickled in diluted acid to remove all mill scale. It is then washed and lubricated.

Going to the drawing bench the mandrel rod with the drawing mandrel secured to it is inserted into the tube and the tube is placed on the bench with the pointed end projecting through the die to be gripped by the pulling mechanism. On attaching the gripper to the moving chain the

tube is drawn through the die and over the mandrel, reducing the diameter and wall. The average reduction made at a single pass varies from about 18 per cent. for small tubes up to 25 per cent. for larger ones. Any number of passes can be taken, but after each pass the tube must be annealed and again "pickled" and lubricated.

These are the standard operations in the production of seamless tubes. No mention has been made of the operations for forming and finishing the tubes, produced by the cupping process, been described, as the tonnage produced by this process is comparatively small.—"Page's Engineering Weekly."

VISCOSITY AND ITS RELATION TO LUBRICATING VALUE.

By Alan E. Flowers,

Professor of Electrical Engineering, Ohio State University.

SYNOPSIS.—The article describes what is meant by viscosity and the effect of that quality upon the "body" and lubricating value of an oil. The moving parts of a bearing are separated by a cushion of oil, the film adjacent to each surface sticking to that surface. The motion is thus between particles of the fluid, and the bearing obeys the laws of fluid friction. This gives the definitions that "fluid lubrication" is the separation of solid surfaces by means of a fluid of suitable viscosity, and that "body" is this ability to be dragged between and so to lubricate surfaces. The resistances increase with the viscosity and the velocity, but decrease with increased clearance.

Practicing and operating engineers have in the past quite generally believed that viscosity was an abstract term related only to some obscure, scientific way of describing materials and having no relation to real lubricating value in everyday service. Even investigators were cautious about claiming any relation between viscosity, friction and power loss due to friction. Happily, this situation is gradually being changed, owing:

First, to a better analysis of the readings which have in the past been accepted as measuring viscosity; and second, to a clearer recognition of all the factors that affect the friction between two rubbing surfaces and a more exact measurement of these values.

It is not surprising that a relation between viscosity and friction could not be proved earlier, since we did not have, till quite recently, a correct knowledge of the real, or "absolute," viscosity of the lubricants, due to the errors of the flow-type viscosimeters in use, or a recognition and measurement of the effect of speed and clearance between bearing and journal, on the amount of friction. The labors of many different investigators have contributed to the solution of these difficulties, and a few of these are mentioned in the appended list of references. In the papers and in the discussion of the reports presented to the American Society for Testing Materials in 1914 and 1915, will be found additional information on viscosity.

DIFFERENT KINDS OF FRICTION.

It is the purpose to present here the elementary, fundamental principles in the clearest and simplest way and in such fashion as will make the knowledge immediately useful. At the same time it is proposed to lay a correct foundation for any further study of the subject.

In order, therefore, to make perfectly clear what is meant by the term viscosity, let us consider for a moment the different kinds of resistance to motion.

There is first the resistance to the sliding of one solid body on another. Subject to slight modifications and variations, the general laws for friction

of this kind are, that to keep up such a sliding motion: (1) The frictional force required is independent of the area of contact; (2) the frictional force is independent of the velocity (except that when starting, the force required to start the motion may be nearly double that required to continue it); (3) the frictional force increases directly as the total pressure between the rubbing surfaces.

These laws for the frictional force may be repeated in a much more convenient mathematical form as follows:

$$F = cP$$

where F is the force required for motion or is the frictional force, P is the total pressure between the rubbing surfaces, both of these quantities being expressed in the same units, and the "constant" c is the so-called friction coefficient.

Contrary to what might be expected and probably also to what is generally believed, well-lubricated bearings are not subject to these relations except when being started from rest or when under excessive loads, because when running the motion carries lubricant along and forces it between the surfaces, actually separating them so that the journal runs on a liquid cushion. Naturally, then, the behavior of a bearing is governed by the laws of fluid motion. Only imperfectly lubricated bearings or those with rough surfaces or those where the journal is warped or bent allow metallic contact and follow to any extent the laws of friction between solids. Someone has said that "there are no laws for imperfectly lubricated surfaces." This is putting the case rather strongly, but it emphasizes strikingly the possible dangers and the probability of large and unexpected variations of friction, with consequent excessive heating.

WELL-LUBRICATED BEARING FRICTION.

The laws of fluid motion or of motion of a solid through a fluid must, then, be the ones to be considered in well-lubricated bearing friction, and it is necessary to form a clear idea of what we mean by a fluid as distinguished from a solid. One might say that the thing that distinguishes a fluid from a solid is its fluidity, meaning by this the ability of any part to move freely with regard to any other part. In other words, the parts of a fluid yield to the slightest forces and motion results, no matter how small the force may be, while a solid resists and no motion takes place till the force exceeds a certain fixed value. Small forces, less than the so-called elastic limit, may twist a solid, but when the force ceases the solid resumes its original shape. The only limitation to this fundamental distinction between solids and fluids is that some materials possess plasticity; that is, they resist small forces, particularly those suddenly applied, but yield under larger forces or under the continued application of even small forces. Beautiful examples of these plastic materials are asphalts, pitches and greases. These materials acts as solids under small forces and as fluids under large forces or continuously applied forces.

It is a matter of common observation that some fluids flow more freely than others; that is, there is greater freedom for the motion of one part through the rest of the fluid. In the more exact terms of the engineer, a greater amount of motion results from the application of the same amount of force. The converse or reciprocal of this property of fluidity, which might be called the resistance to relative motion, is the property to which the name viscosity is given. Thus we have the paradox of defining fluids by their viscosity, or their *degree of lack of fluidity*. Internal friction is not a good substitute for the term viscosity, because it implies "rubbing," as of solids, and absence of motion till the force exceeds the value needed to tear the particles along.

It is plain that when the parts of a fluid move there is this property of viscosity-resisting motion, and we speak of this as the viscous resistance.

VISCOUS RESISTANCE OF A FLUID.

It is quite natural to ask whether the viscous resistance that is due to the action of parts of the fluid on one another can account for all the resistance when a solid moves through a fluid or a fluid flows along a wall, as when a fluid flows through a pipe or a journal turns on its cushion of oil. This is a legitimate question and can be answered only by experiment.

There are two things to be looked for. The first is to determine whether there is any slippage, or relative motion between the solid wall and those particels of the fluid nearest the wall. Here the experimental answer is very positive and final. It is, that except in rare cases the particles of the fluid in actual contact with the walls adhere so strongly to the walls that there is no slippage or relative motion of the solid and this part of the fluid, so that the solid is surrounded by, and carries with it, a small portion of the fluid, and consequently all the relative motion is between fluid and fluid and all the resistance is due to the laws of fluid motion. Even the fluids that do not stick to and wet the walls, as in the case of mercury and glass, behave in this way to such an extent that only a negligible fraction of the resistance can be due to slippage. For one particular case that was investigated, where the flow of mercury through small tubes was measured, not even one five-hundredth of the resistance could have been ascribed to friction between fluid and solid due to slippage of the fluid along the wall.

The second thing to be looked for is the effect of the inertia of the particles of the fluid when they are brought from standstill to some appreciable velocity or when hurled out of the way of a moving body or when the particles of the fluid are hurled hither and thither in eddies and turbulent motions in those cases where there is plenty of room in the vessel or between the walls of the tube or where there is a large clearance between the two bearing surfaces.

WHY TESTS LACK CONSISTENCY.

This has been the stumbling block that has destroyed much of the value of otherwise good work, for experiment answers that "it all depends." In other words, for high velocities, for rough walls, for large tubes, passages and clearances, for dense fluids and for materials of very low viscosity, a large part of the resistance to motion may be due to inertia resistances.

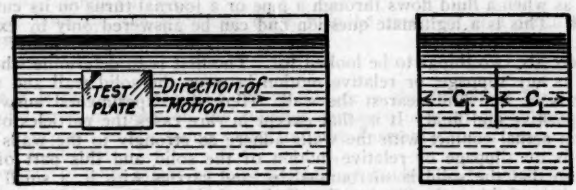
If, then, we keep the velocity low enough or use such shapes that the fluid is not hurled about in eddies, we may feel sure that all the resistance to motion is due to the lack of freedom of motion of one particle of the fluid on another; that is, to the property of viscosity of the fluid.

By observing that the slowing down of a body oscillating slowly to and fro when immersed in a fluid is the same at one depth of immersion as at another, it has been shown that viscous resistance is independent of the pressure on the fluid (except for enormous pressures).

It is evident that the amount of resistance would be greater for a large area and for higher relative velocities, and careful experiment has abundantly confirmed these relations. Just what is meant by relative velocity is made more clear by considering specific cases.

Suppose we had a long, deep, narrow vessel and put a thin plate parallel to and midway between the side walls, as in the accompanying sketch. It is evident from the foregoing that when the test plate is moved a film of fluid adheres to the plate and moves with it at some velocity, which we may designate at V_1, while a film next the vessel walls adheres to them and so is stationary. Intermediate portions would move at some intermediate velocity. From this it would appear that the total velocity of fluid on fluid is equal to V_1, the velocity of the test plate, and that this velocity is distributed over the distance c_1, separating the test plate and wall so that

the average relative velocity would be v_1/c_1. In a wider vessel, where the separation could be made twice as great, the same velocity V_1 would be distributed over $2c_1$, twice as great a distance, and the average relative velocity would be $V_1/2c_1$ or would be half as great as before. Under



ILLUSTRATING EFFECT OF CLEARANCE.

these conditions the test plate would evidently move more easily, and the greater ease of movement is directly the result of the lower relative velocity. For clearances small enough to make feasible the use of average velocities, the resistance to motion would vary directly with the relative velocity. These relations may be stated more concisely as follows:

$$R_v \textit{ varies with } A\frac{V}{c}$$

where A is the surface area, V is the velocity, c is the clearance.

If a test were made with a plate of unit area or the resistance per unit area determined from a test for a case where the relative velocity was unity, a certain value for the resistance would result and the value of this unit of resistance would be characteristic of the fluid used—in particular a characteristic of the lack of fluidity of the fluid; and if the value of this unit resistance be designated by r, we have a means of calculating the resistance for any case by multiplying the resistance per unit area and per unit relative velocity by the total area and by the relative velocity.

Mathematically, we may write the expression for the viscous resistance as:

$$R_v = rA\left(\frac{V}{c}\right)$$

where

R_v = Viscous resisting force for any case;
r = "Viscosity" or the viscous resisting force per unit area and per unit relative velocity;
A = Surface area in motion;
$\left(\frac{V}{c}\right)$ = Relative velocity.

The general property of viscosity is, then, the lack of freedom of flow; and put more precisely in engineering terms, the viscosity is the resisting force per unit area per unit relative velocity.

When clearances are large and the actual velocity of the intermediate portions of the fluid cannot well be represented by the average relative velocity $\left(\frac{V}{c}\right)$, a better form of statement is to use the calculus form and write $\left(\frac{dv}{dc}\right)$, meaning by this the ratio of the difference of the velocities of two closely adjacent very thin sheets of the fluid to the difference of their positions (or their distance apart).

WHY PRESENT VISCOSIMETERS FAIL.

The reason all the present flow-type viscosimeters give wrong results and so have made impossible a true correlation between viscosity and lubricating value is that they are standardized with water, which flows out so rapidly that about five-sixths of the resistance is due to inertia effects and eddies and only the small remainder is due to the viscosity. In these instruments, each of which consists of a vessel with a short, narrow outlet in the bottom, through which the fluid passes, the time required for a given quantity to pass out, determined by means of a stop-watch (the outflow time), is taken as a measure of the viscosity. When standardized with water, an instrument such as the Saybolt requires, say, 28.5 seconds for the outflow of water at 20 degrees C. or 68 degrees F. Some oil may require twice as long, or 57 seconds, but such an oil would be really nearly ten times as viscous, instead of just twice. The time it ought to take for water to flow out would be about 5½ seconds if viscosity alone determined its flow. Another way of putting it is to state that if you could test a material of the same density as water so that the inertia effects would be the same, but having no viscosity at all, that is, infinite fluidity, the outflow time for such a material would not be zero, but would be almost exactly 27 seconds; that is, it would get out only 1½ seconds quicker than water.

Some years ago Ubbelohde gave the form and the constants for calculating the absolute viscosity from the readings taken with the Engler viscosimeter. More recently Meissner (6) has compared the readings of the Engler, the Redwood and the Saybolt viscosimeters. These results are restated in the written discussion of the Report of Committee D2 on Standard Tests for Lubricants at the 1915 convention of the American Society for Testing Materials, along with additional expressions, so that the absolute viscosity may be calculated from the readings as well as the readings corresponding to any viscosity, provided the density is also measured, since these instruments give readings that depend on both the viscosity and density. The following symbols may be used:

E = Ratio of the outflow time to the water time. The water time is taken at 20 degrees C. or 68 degrees F.
t = Outflow time, in seconds.
n = Absolute viscosity.
d = Density or specific gravity.

For the Engler viscosimeter, whose water time is 51.0 sec.,

$$\frac{n}{d} = 0.08019\,E - \frac{0.07013}{E}$$

$$\frac{n}{d} = 0.0015723\,t - \frac{3.576}{t}$$

$$t = 318.0\left(\frac{n}{d}\right)\left(1 + \sqrt{1 + \frac{0.0225}{\left(\frac{n}{d}\right)^2}}\right)$$

For the Redwood viscosimeter, No. 1,263, which was tested and found to have a water time of 26.47 seconds,

$$\frac{n}{d} = 0.002631\,t - \frac{1.5770}{t}$$

$$t = 190.0\left(\frac{n}{d}\right)\left(1 + \sqrt{1 + \frac{0.01660}{\left(\frac{n}{d}\right)^2}}\right)$$

For the Saybolt viscosimeter, No. 752, which was tested and found to have a water time of 28.57 seconds,

$$\frac{n}{d} = 0.0021863\, t - \frac{1.4968}{t}$$

$$t = 228.7 \left(\frac{n}{d}\right)\left[1 + \sqrt{1 + \frac{0.01309}{\left(\frac{n}{d}\right)^2}}\right]$$

As shown by these expressions for the calculation of the absolute viscosity, the values for the materials that are at all low in viscosity, such as water, are given by the differences of two relatively large quantities, so that the error of the determination is greatly increased. The errors due to inertia effects and eddies cease more or less abruptly, an effect not taken account of in these equations, and so for some readings the error in the final result will differ from that given by these equations.

The situation would not be so bad if all the oils tested were very thick or viscous and if they did not decrease in viscosity so greatly when warmed to the high temperatures to which they are subjected in service.

AN IMPORTANT DEMONSTRATION.

In spite of this handicap Ubbelohde (7), after compiling and comparing the results of former investigations, has shown that oils from different crude-oil fields and prepared by different methods of refining but having the same absolute viscosity give the same friction coefficient under similar test conditions.

If Ubbelohde's conclusion is substantiated, a very great advance will have been scored and it will be possible to duplicate any friction result by duplicating the viscosity of the lubricant.

It is very plain, too, that as long as it is possible to keep a continuous sheet of lubricant between the bearing surfaces and to keep the same separation of the stationary and moving parts, the lower the viscosity the lower the friction loss.

The consideration that sets a lower limit to the viscosity and therefore to the friction-force coefficient is the speed and the load pressure on the bearing. The moving part carries the viscous lubricant along with it to the surfaces to be separated (provided no sharp edges are allowed to scrape the lubricant off). Anyone who has stuck his finger into thick molasses and pulled out half a cupful clinging to his finger will appreciate that it is its high viscosity, or lack of freedom of flow, that allows him to pull out so much molasses, while he would be able to bring out only a very small amount of water in the same way on account of the low viscosity of water and the ease with which all but the film wetting the finger would flow off the finger as it was withdrawn. Just so the viscosity of a lubricant helps in lubrication by getting the lubricant to the place where it is most needed. It is this property of viscous materials that allows them to get between surfaces to which the name "body" has been applied.

Hersey (3) has stated that the carrying power of a bearing increases directly with increase of viscosity and also increases with increase of speed, but reaches a limiting value, so that further increase of speed causes no increase of carrying capacity.

SUMMARY OF RESULTS.

We might summarize these results by saying that since the lubricant adheres to the walls, the relative movement occurs between the different parts of the lubricant and the resistance obeys the laws of the resistance of fluid motion.

With smooth, narrow passages, or low velocities, so that turbulence is avoided, the principal part of the resistance is due to the lack of fluidity; that is, to the viscosity of the lubricant.

The viscous resistance for different cases increases with the viscosity and the relative velocity; that is, it increases with the total velocity, but decreases with increased clearance.

In general, the lower the viscosity the lower the friction, but since the stationary and moving surfaces should have as much lubricant between them as possible and since a thin film of the lubricant always adheres to the walls, it is necessary to have sufficient viscosity for the film attached to the moving part to drag along enough more lubricant to properly separate (that is, "lubricate") the surfaces, and the "body" of a lubricant is its ability to be dragged between surfaces.

Since a higher speed would carry along the lubricant better, it is possible to get satisfactory lubrication at high speeds with low-viscosity materials. This explains the need for low-viscosity lubricants for high speeds and high-viscosity lubricants for low speeds.

When the pressure between the parts is low, it is easy to get and keep lubricants between the surfaces and low-viscosity lubricants may be used, but if the pressure is high, tending to squeeze out the lubricant, high-viscosity lubricants must be used.

References—(1) Petroff, "Baumaterialkunde," Vol. 4, p. 269, 1899. (2) Sommerfeld, "Physikalische Zeitschrift," Vol. 4, p. 770, 1903. (3) Hersey, "Journal of the Washington Academy of Sciences," Vol. 4, No. 19, Nov. 19, 1914; "Journal of the American Society of Mechanical Engineers," September, 1915, Vol. 37, p. 534. (4) Ubbelohde, "Chemiker Zeitung," 1907, Vol. 31, p. 38. (5) Grüneisen. "Wissenschaftliche Abhandlungen der Physikalisch-Technischen Reichsanstalt," Vol. 4, No. 2, p. 154, 1904. (6) Meissner, "Chemische Revue," Vol. 12, p. 30, 1912. (7) Ubbelohde, "Petroleum," Vol. 7, p. 773, 1912. (8) American Society for Testing Materials, Technical Papers and Reports, 1914 and 1915.—"Power."

NAVY HAS LARGEST EXPERIMENTAL WIND TUNNEL.

By William McEntee,

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The large experimental wind tunnel that the Navy Department has established in the Washington Navy Yard at the experimental-model basin has now been in operation about a year. The tunnel is the largest in the world, having a section 8 feet square at the point where the models are placed for testing. In addition to the advantage gained by the size it is possible with a 500-H.P. motor-driven fan to get wind speeds up to 75 miles per hour, which permits experiments being made at real flying speeds.

The tunnel consists of a closed circuit shaped like the link of a chain, as shown in Fig. 1. The 500-H.P. top-horizontal discharge fan is of the corrugated-paddle type, with an inlet diameter of 11 feet 2 inches and a discharge duct 7 feet 6 inches by 9 feet, and is placed at one end of the link. At the other end, where the air straightens out before flowing through the experimental chamber, are the baffles, which are necessary to remove the eddies and to control the uniformity of the speed. These baffles consist of 64 cells, each 1 foot square and 8 feet long. Each cell is provided with its own damper, so that the velocity of the air in any one section can be controlled. At the experimental chamber in the vicinity of which aeroplane wings or models are tested the maximum variation from uniform flow is about 2 per cent.

The tunnel is built of wood; frames are spaced about 3 feet on centers and are sheathed on the inside with ⅞-inch tongued-and-grooved sheathing

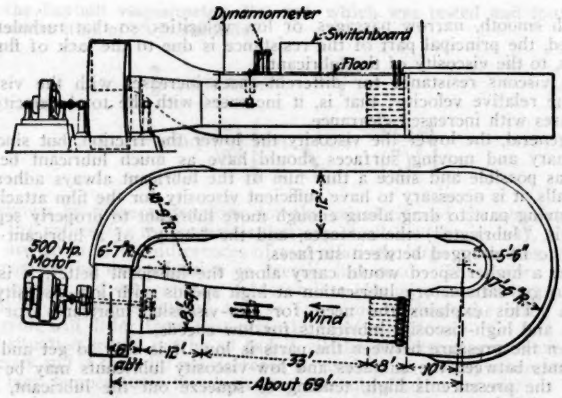

FIG. 1.—LAYOUT OF U. S. NAVY WIND TUNNEL.

laid in two thicknesses in the direction of the air current and with building paper placed between the two layers. The necessary curvature is obtained by bending the sheathing, the whole of which is blind nailed.

The fan is driven by a 250-volt, 500-H.P. direct-current motor, arranged with Ward-Leonard and auxiliary-field speed control, so that any desired speed up to about 200 r.p.m., which corresponds to a wind speed of 75 miles

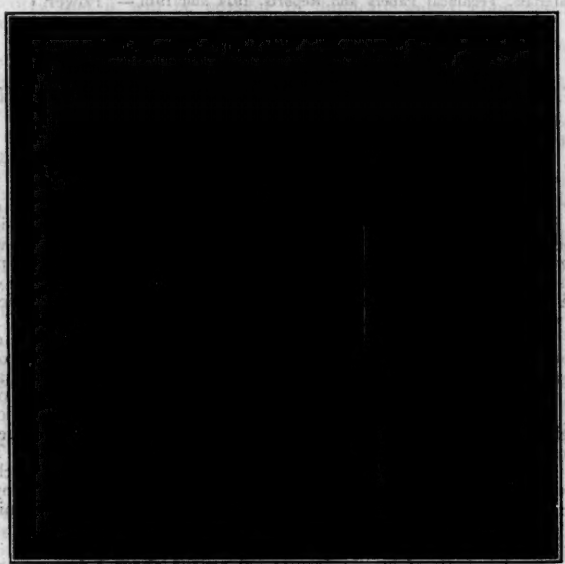

FIG. 2.—MODEL AEROPLANE IN TESTING POSITION, NAVY WIND TUNNEL.

per hour, may be obtained. At the discharge side of the fan are located 12 pitot tubes, which lead to an integrating manometer that gives the average velocity of discharge. This velocity has been calibrated against the velocity obtained at the section in the experimental chamber where the aeroplane or other model is placed, so that any desired velocity may be obtained with precision at that point without having any pitot tubes or obstructions other than the model being tested. In other words, by calibration the velocity of discharge may be found, and this bears a certain constant ratio to the velocity at the experimental section.

The velocities were determined by pitot tubes that were checked with those used in the aerodynamical laboratory of the Massachusetts Institute of Technology and in the National Physical Laboratory, England.

INVESTIGATIONS ALREADY MADE.

Among recent investigations of interest made at the navy wind tunnel was the determination of the coefficient of air friction for various aeroplane and balloon fabrics. Tests have been made on the new dirigible building for the Navy Department and on models of naval aeroplanes both building and projected. A number of tests have also been made for private concerns. In carrying out experiments for private parties the same practice is followed as in the case of tests of ship models; that is, the actual cost of doing the work is charged in each case. On account of the large size of the tunnel, it is possible to test comparatively large models of aeroplanes with widths up to 36 inches.

In Fig. 2 is shown the arrangement of the model of an aeroplane when being tested. The model is carried by a steel spindle that extends up through the top of the tunnel to the weighing balance, which is placed overhead. For about two-thirds of the length in the tunnel the spindle is covered by a mask of stream-line form. This mask is secured to the ceiling of the tunnel and reduces the force acting on the spindle itself, and thus the spindle correction. The weighing balance consists of a weighing scale on the platform principle having three axes—two of them at the same horizontal line, 61 inches apart, and the third vertically over one of the first, 48 inches above it. When a model is set at a given angle, the moments acting about each of these three axes are measured by weighing them on the scale. With these data it is possible to compute horizontal and vertical components of the force acting on the model—that is, the drift and lift—and also to compute the line of application of the force. Tests are usually made at speeds of 40 miles per hour. At this speed and at the angle of least resistance an ordinary aeroplane-wing model has a horizontal resistance of something less than 1/10 pound. It is therefore necessary that the balance should be capable of weighing a force with accuracy to about $^{2}/_{1000}$ pound.

The large size of the tunnel makes it possible to test full-sized radiators for aeroplane motors, and comparative tests have recently been made on several types, both as to air resistance and cooling capacity.

The wind tunnel has also been used for certain other tests that are not directly connected with aeronautics, such, for example, as the determination of the influence of form and dimensions on the size of ventilating cowls for use on ships. These tests have shown that it is not necessary to exceed certain dimensions, which are less than have heretofore been used in many cases. It is proposed shortly to obtain the wind resistance of a large battleship, this being an element of a ship's resistance that has not previously been accurately measured.

In conjunction with the wind tunnel the model basin is used for determining the best form of floats for hydroplanes, and there will no doubt be many other cases where the data derived from tests made in the water may be applied directly to aeroplane design.—"Engineering News."

STEAM TURBINE BLADING.

The blading of a steam turbine is so vital that if it becomes damaged or eroded the efficiency of the unit will be seriously affected. A failure of even one blade may completely wreck the unit. Therefore strength and the property of resisting erosion and corrosion are paramount.

Builders had to feel their way with their first turbines until they learned by experience the performance of various materials. In this way they developed blading that gives excellent service in their respective machines. But the constant demands for increased size of unit, for better economy and for turbines to operate with high steam pressure and high superheat will make it necessary to seek better blading materials than those now in use. The increases in size and economy can be secured partially by increased blade speeds. This results in increased centrifugal speeds and correspondingly increased working stresses in the blades. Increased steam pressure and superheat also increase capacity. Much of the blading material that has proved satisfactory under moderate temperatures would become unsafe under the temperatures corresponding to very high pressures and superheat. When such blades are subject to continuous high temperatures, they become annealed and lose much of their strength. Even when not subject to annealing, all metals tend to decrease in strength when heated, and the strength of bronze decreases more rapidly than that of steel.

A paper was recently read before the Institute of Metals, dealing with alloys for high-speed superheated-steam turbine blading. The author discussed the various materials used and told how each behaved. He showed that brass alloys of copper and zinc are unsatisfactory for highly superheated steam, as they anneal rapidly and become too ductile and weak. Copper was once used extensively in England for high-pressure blading, but is unsuited for high temperatures. He also found that copper-nickel alloys, while much better than the brasses, are also subject to softening. His investigations led him to conclude that the best alloy blading for superheat would be a phosphor-bronze or an alloy of copper-nickel-aluminum.

In Europe a 5-per cent. nickel-steel was found to work satisfactorily for moderately high superheat. A French engineer has pointed out that if pure nickel is alloyed with soft steel from an electric furnace to give a 5-per cent. nickel-steel, this gives excellent service. Some builders use soft steel with little carbon, silicon or manganese, but this material rusts easily. It is thus evident that there is a wide field open for the chemist and metallurgist to produce a blading material that will successfully withstand high superheat and high rotational speed and will not corrode in an atmosphere of wet steam.

Higher blade speeds permit higher steam speeds. Improved efficiency, together with higher steam pressures and superheat, will probbaly produce steam in the last rows of blades as wet as, if not wetter than, is now common. Wet steam traveling at high speed in the exhaust end corrodes and erodes the metal. Therefore low-pressure blading must be practically non-corrosive. This is more necessary in the case of impulse types such as the Curtis or Rateau than in the Parsons, or reaction, blading, for the inlet angle of the blades is a factor of much importance in the first two types. In many turbines of these types now in use, the low-pressure blading has to be renewed periodically because of the eating away of the material by the wet steam.

What might be called the "sand-blast" action of the steam also has to be considered. The high velocity of the steam tends to tear away particles from the first rows of blades, and these, together with the moisture in the steam, produce a scouring action on the low-pressure blading. Furthermore, many users of turbines have noticed that deposits of lime and soda occur on the backs of some of the low-pressure blades, and evidently

came over with the superheated steam, but were not deposited until the steam became wet. In one case examination failed to show scale or other material in the superheaters. The only logical explanation is that the matter had been carried over in wet steam. As steam pressures and temperatures increase, the character of the feed water and the condition of the fluids in the boilers become matters of importance. This has been recognized at the Connor's Creek plant of the Detroit Edison Co., where distilled water is used for boiler-feed purposes.

The demand for increased size of units and increased efficiency of steam turbines has increased the problems before builders in an almost incredible degree. In justice to them it is only fair that patience and forbearance be shown them while they are finding materials that will serve under these new conditions.—"Power."

GYRO STABILIZERS.

The idea of steadying vessels by means of gyroscopes is not new. Herr Schlick, years ago, made experiments on a small vessel, the *Seebar*, which demonstrated their utility in reducing rolling. Why greater progress has not been made towards a more general adoption of such means for producing steadiness it is difficult to say. Frahm's anti-rolling tanks were preceded by many years by the rolling tanks in the old *Inflexible*, yet no commercial use was made of the idea until a comparatively recent date. There is every likelihood that the Sperry gyro stabilizer will prove commercially successful and be adopted in many passenger and war vessels in which steadiness is of great importance. Sperry's experiments on the *Widgeon*, which had the reputation of being a very free roller, displayed the steadying effect of his apparatus. With equipment which weighed about 1 per cent. of the displacement, the vessel's rolling was reduced in a particular sea from 40 to 3 degrees. The weight of the gyroscope does not bear a constant ratio to that of the ship, as waves of given magnitude have more influence on small ships than they have on large ones, and therefore require greater roll-quenching power to be installed.—"Shipbuilding and Shipping Record."

UNITED STATES NAVAL VESSELS UNDER CONSTRUCTION.

DEGREE OF COMPLETION.

No.	Vessel.	Building yard.	Engines.	No. shafts.	Speed, knots.	Percentage machinery completed 1916.		Percentage hull completed May 1, 1916.	
						Apr. 1	May 1	Total.	On ship.
	BATTLESHIPS:								
38	*Pennsylvania*	Newport News Co	Cur. trb. grd. cr	4	21	96.08	97.00	99.3	99.3
39	*Arizona*	Navy Yard, N. Y.	Pars. trb. grd. cr.	4	21	80.98	84.10	89.9	89.2
40	*New Mexico*	Navy Yard, N. Y.	Electric	4	21	5.28	5.29	26.1	19.5
41	*Mississippi*	Newport News Co	Cur. trb. grd. cr	4	21	32.25	34.78	43.9	32.9
42	*Idaho*	New York S. Co	Pars. trb. grd. cr.	4	21	44.51	48.43	59.1	55.3
43	*Tennessee*	Navy Yard, N. Y.	Electric	4	21	...	...	...	...
44	*California*	Navy Yard, Mare Isl'd	Electric	4	21	...	...	...	...
	DESTROYERS:								
62	*Wainwright*	New York S. Co	Pars. trb. grd. cr.	2	29.5	99.66	99.74	99.9	99.9
63	*Sampson*	Fore River S. Co	Curtis trb. grd. cr.	2	29.5	93.51	96.27	87.7	87.1
64	*Rowan*	Fore River S. Co	Curtis trb. grd. cr.	2	29.5	88.01	90.15	82.1	81.2
65	*Davis*	Bath Iron Works	Pars. trb. grd. cr.	2	30	86.97	89.04	79.6	78.8
66	*Allen*	Bath Iron Works	Pars. trb. grd. cr.	2	30	85.46	87.31	77.0	75.9
67	*Wilkes*	Wm. Cramp & Sons	Pars. trb. grd. cr.	2	29.5	66.37	77.35	63.4	61.7
68	*Shaw*	Navy Yard, Mare Isl'd	Pars. trb. grd. cr.	2	29.5	54.63	56.85	33.6	31.4
69	.....	Navy Yard, Mare Isl'd	Curtis trb. gearing	2	32	2.90	3.00	...	...
70	.....	Navy Yard, Norfolk	Pars. trb. gearing.	2	32	...	...	...	...
71	.....	Seattle Con. & D. D. Co.	Pars. trb. gearing.	2	30	...	...	...	...
72	.....	Wm. Cramp & Sons	Pars. trb. grd. cr.	3	30	...	2.94	...	...
73	.....	Wm. Cramp & Sons	Pars. trb. grd. cr.	3	30	...	3.00	...	...
74	.....	Bath Iron Works	Pars. trb. gearing	2	32	3.14	8.03	9.1	2.7
	FUEL SHIPS:								
14	*Maumee*	Navy Yard, Mare Isl'd	Diesel	2	14	89.06	89.73	99.7	99.7
15	*Cuyama*	Navy Yard, Mare Isl'd	Reciprocating	2	14	19.70	24.89	50.3	46.0
	DESTROYER TENDER:								
	Transport	Navy Yard, Phila	Reciprocating	2	14	35.84	38 00	58.9	56.7
	Supply ship	Navy Yard, Boston	Reciprocating	2	14	25.00	25.92	60.3	58.2
	TUGS:								
17	*Wando*	Navy Yard, Charleston	Recip. oil fuel	1	11	31.70	35.62	...	...
18	*Pocahontas*	Navy Yard, Norfolk	Recip. oil fuel	1	11	50.91	59.17	...	...
	Wave	Navy Yard, Charleston	.....	...	...	57.70	61.73	...	...
	SUBMARINES:								
31	*G-3*	Navy Yard, N. Y.	Diesel-Sulzer	2	14	93.00	93.00	89.5	89.3
41	*L-2*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	98.41	98.41	99.1	99.1
44	*L-3*	Lake Co., Bridgeport	Diesel-Sulzer	2	14	61.00	76.00	86.1	84.9
45	*L-6*	Lake, Long Beach, Cal.	Diesel-Sulzer	2	14	50.68	65.59	82.4	79.5
46	*L-7*	Lake, Long Beach, Cal.	Diesel-Sulzer	2	14	46.57	62.90	81.2	78.3
47	*M-1*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	97.27	97.29	96.0	96.0
48	*L-8*	Navy Yard, Portsmouth	Diesel-Sulzer	2	14	21.53	26.43	85.7	83.7
49	*L-9*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	95 48	95.68	94.2	94.2
50	*L-10*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	95 27	95.67	92.7	92.7
51	*L-11*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	94.63	95.14	90.4	90.1
52	*Schley*	Elec. Boat Co., Quincy	Diesel-New Lond	2	20	6.75	12.32	12.4	5.3
53	*N-1*	Elec. Boat Co., Seattle	Diesel-New Lond	2	13	60.68	69.71	56.1	46.2
54	*N-2*	Elec. Boat Co., Seattle	Diesel-New Lond	2	13	47.33	69.89	56.1	46.2
55	*N-3*	Elec. Boat Co., Seattle	Diesel-New Lond	2	13	45.35	49.68	56.1	46.2
56	*N-4*	Lake Co., Bridgeport	Diesel-Sulzer	2	13	9.90	12.50	64.2	58.7
57	*N-5*	Lake Co., Bridgeport	Diesel-Sulzer	2	13	9.90	12.50	63.2	57.6
58	*N-6*	Lake Co., Bridgeport	Diesel-Sulzer	2	13	9.90	12.50	61.4	55.7
59	*N-7*	Lake Co., Bridgeport	Diesel-Sulzer	2	13	9.90	12 50	61.5	55.4
62	*O-1*	Navy Yard, Portsmouth	Diesel-New Lond	2	14	...	...	...	...
63	*O-2*	Navy Yard, P. S.	Diesel-New Lond	2	14	...	...	...	...
64	*O-3*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	...	0.10	...	...
65	*O-4*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	...	0.10	...	...
66	*O-5*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	...	0.10	...	...
67	*O-6*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	...	0.10	...	...
68	*O-7*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	...	0.10	...	...
69	*O-8*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	...	0.10	...	...
70	*O-9*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	...	0.10	...	...
71	*O-10*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	...	0.10	...	...
72	*O-11*	Lake Co., Bridgeport	Diesel-Sulzer	2	14	0.08	0.10	11.2	5.9
73	*O-12*	Lake Co., Bridgeport	Diesel-Sulzer	2	14	0.08	0.10	10.9	5.8
74	*O-13*	Lake Co., Bridgeport	Diesel-Sulzer	2	14	0.08	0.10	9.9	5.8
75	*O-14*	Cal. S. B. Co., L. B.	Diesel-Sulzer	2	14	1.66	2.29	7.5	5.0
76	*O-15*	Cal. S. B. Co., L. B.	Diesel-Sulzer	2	14	1 68	2.84	7.0	5.0
77	*O-16*	Cal. S. B. Co., L. B.	Diesel-Sulzer	2	14	1.69	2.26	6.8	4.9

BOOKS RECEIVED.

OIL FUEL. By ERNEST H. PEABODY.—A treatise on oil fuel presented at the International Engineering Congress, San Francisco, California. Treats of Oil Discovery and Production, Distributing Points and Storage at Shore Stations, Comparative Cost of Oil Fuel, Properties of Oil, Storage on Board Ship, Methods of Burning, etc. This valuable pamphlet will be welcomed, as it has everything needful to make it a standard for engineers on the subject of oil fuel.

INDUSTRIAL USES OF FUEL OIL. By F. B. DUNN.—An excellent treatise for the use of practical men. It is a practical exposition of the use of fuel oil for industrial purposes. Contains eighteen chapters with numerous illustrations, and treats of oil fuel in all its phases. The language is simple, the text large and clear, and the numerous illustrations, which are a host of information in themselves, are beautifully executed. TECHNICAL PUBLISHING COMPANY, 618 Mission Street, San Francisco, Cal.

FUNDAMENTALS OF MILITARY SERVICE. By CAPTAIN LINCOLN C. ANDREWS, U. S. Cavalry.—A text book and system of training that will enable each citizen to know enough about the military service and its needs to give him an intelligent opinion thereon, and aid him in preparing himself to meet the individual responsibilities as a citizen soldier. Contains twenty-five chapters treating of Our Military Service, Psychology of the Service, Military Training, Organization, Infantry Drill, Cavalry, Field Artillery, Coast Artillery, The Engineer Corps, Signal Corps, Tactical Rules, Regulations, Small-Arm Firing, etc. J. B. LIPPINCOTT COMPANY, Philadelphia.

EFFICIENCY IN THE SCREW PROPELLER.—A pamphlet issued by the AMERICAN SCREW PROPELLER COMPANY, of Philadelphia, Pa. This company designs propellers by the methods of design brought out by Captain Charles W. Dyson, U. S. N., as the fruit of fourteen years of close study of this subject. These methods have been published from time to time in the pages of this JOURNAL.

The DIESEL ENGINE IN PRACTICE. By J. E. MEGSON and H. S. JONES.—Contains twelve chapters and gives the benefit of many years of practical experience in operating the Diesel engine. Includes Historical Outline, Basis of Operation, Experience with Earlier Installations, Fuel Oil, Operation and Care of Engines, Modern Engines, Diesel Applied to to Marine Purposes, etc. TECHNICAL PUBLISHING COMPANY, 618 Mission Street, San Francisco, Cal.

ASSOCIATION NOTES.

The following members and associates have joined the Society since the publication of the last JOURNAL.

MEMBERS.

Anderson, Lorain, Lieutenant, U. S. N.
Barry, James R., Lieutenant, U. S. N.
Bowman, Mark C., Lieutenant, U. S. N.
Crenshaw, Edmund A., Lieutenant, U. S. N.
Murray, George D., Lieutenant, U. S. N.
Parrott, George F., Jr., Lieutenant, U. S. N.
Poteet, Fred H., Lieutenant, U. S. N.
Rose, Solon E., Ensign, U. S. N.
Saxer, John J., Lieutenant, U. S. N.
Schlossbach, Isaac, Ensign, U. S. N.
Tisdale, George M., Ensign, U. S. N.

ASSOCIATES.

Cone, J. D., Cameron Steam Pump Works, 11 Broadway, New York City.
Frauenfelder, J. Barraja, Chief Engineer, The Lake Torpedo Boat Company, Bridgeport, Conn.
McIlvain, William M., Lieutenant, U. S. M. C.
Schiller, Peter G., Room 1616, 165 Broadway, New York City.
Turner, Clarence Porter, 8 Front Street, Schenectady, N. Y.

THE ANNUAL BANQUET of the Society was held at the Army and Navy Club in the City of Washington, D. C., on the evening of March 25, 1916. About 200 members and guests

were present, among whom were several members of both branches of Congress. The occasion was a most enjoyable one. Captain J. S. McKean, U. S. Navy, acted as toastmaster and the following gentlemen responded to toasts:

Hon. Josephus Daniels, Secretary of the Navy.
Senator James D. Phelan.
Hon. Ernest W. Roberts.
Mr. J. W. Powell.
Rear Admiral W. S. Benson, U. S. N.
Mr. William L. Saunders.
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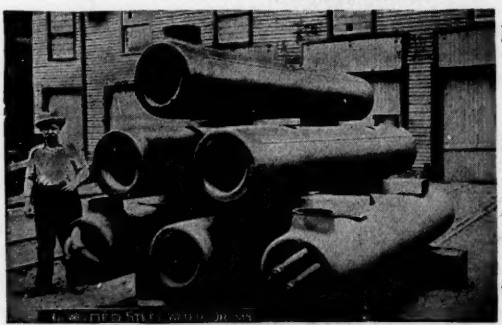

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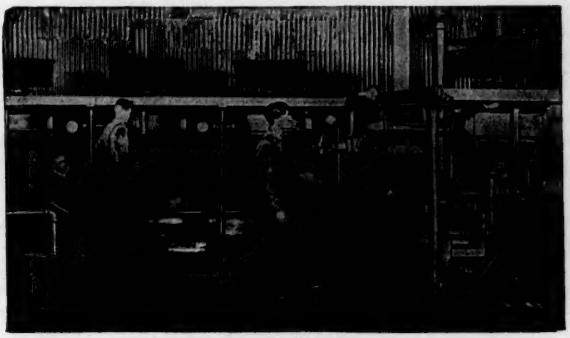

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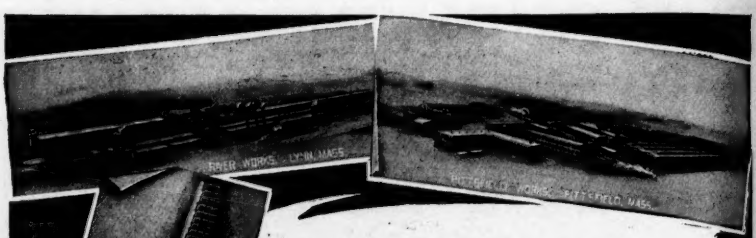

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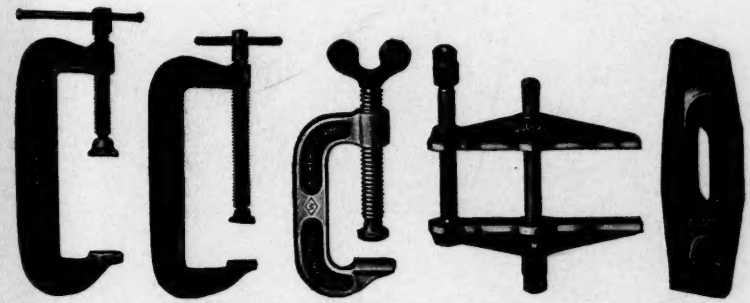

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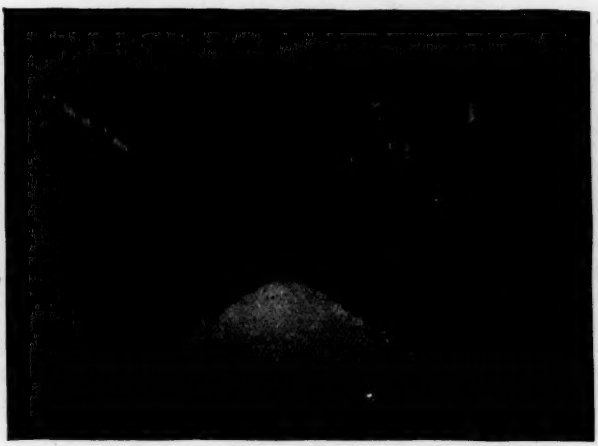

3 MILLION H.P. IN USE

View of 27,000 H.P. Niclausse Boiler Plant fitted with Niclausse Automatic Stokers.

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THE
LOVEKIN ASSISTANT CYLINDERS FOR VALVE GEARS.

The Lovekin Assistant Cylinders are designed for balancing the forces due to **BOTH THE INERTIA AND WEIGHT OF THE VALVE GEAR.** They are, therefore, of great value to engines **FITTED WITH EITHER PISTON OR FLAT SLIDE VALVES.**

Several of the United States Naval Vessels in commission and all those now under construction have their engines fitted with LOVEKIN PATENT ASSISTANT CYLINDERS **TO ALL OF THEIR VALVE GEARS.**

The following Shipbuilding and Engineering Companies have fitted LOVEKIN ASSISTANT CYLINDERS to valve gears on engines built by them:

New York Shipbuilding Co.	Camden, N. J.
Newport News Shipbuilding Co.	Newport News, Va.
Wm. Cramp & Sons Shipbuilding Co.	Philadelphia, Pa.
Fore River Shipbuilding Co.	Quincy, Mass.
American Shipbuilding Co.	Cleveland, Ohio.
Vulcan Works	Stettin, Germany.
Kawasaki Dockyard Co.	Kobe, Japan.
Societa Anonima	Milan, Italy.
N'Odero Co.	Genoa, Italy.

Many of these vessels have run over 250,000 miles without requiring any overhauling of the valve gear. These Assistant Cylinders prevent excessive wear on the eccentric straps or any parts of the valve gear **AND THEREFORE ENABLE THE VALVE GEAR TO BE KEPT IN CORRECT ADJUSTMENT FOR VERY LONG PERIODS SO THAT THE STEAM DISTRIBUTION THROUGH THE VALVES AND PORTS REMAIN AS DESIGNED.**

It is surprising to see the great consideration that is given the flat slide valve frequently used on L.P. Cylinders, few engineers forget to state that "balance cylinders will be used for the L.P. Valve Gears." How many engineers or designers have taken the trouble to find out what a poor device the ordinary balance cylinder is? It can only hold the weight of the valve gear up at one particular receiver pressure. The inertia force which is ever present and varying with the speed of the engine is always the force that causes the trouble and requires a continual expense for upkeep. **THESE INERTIA FORCES ARE PRESENT TO THE SAME EXTENT IN ALL VALVE GEARS IF THEY ARE OF THE SAME WEIGHT,** therefore, why specify the L.P. Valve Gears to be balanced and let the H.P. and I.P. Valve Gears go unbalanced when all valve gears can be balanced so easily by the LOVEKIN ASSISTANT CYLINDER.

THE LOVEKIN ASSISTANT CYLINDER performs the work of neutralizing the forces set up in the valve gear, **DUE TO BOTH THE INERTIA AND WEIGHT** of the same, and, therefore, **SAVE POWER, SAVE WEAR AND TEAR, SAVE OIL, SAVE REPAIR BILLS AND KEEP THE VALVE SETTING CORRECT FOR MUCH LONGER PERIODS THAN IS POSSIBLE WITHOUT THEM.**

The valve gear can be designed much lighter.

Balancing the inertia and gravity forces in the valve gears enables the main engine to be balanced to a far greater extent than is possible where the valve gear forces are not balanced. This was proven quite recently in the case of some high-speed destroyers in the United States Navy, where comparative tests were made with and without LOVEKIN INERTIA CYLINDERS.

NOTE.—NO FORM OF PISTON VALVE OR ORDINARY BALANCE CYLINDER CAN BE DESIGNED TO DO MORE THAN CARRY THE WEIGHT OF THE VALVE GEAR AT ONE CONSTANT PRESSURE IN THE RECEIVER. THEREFORE THE GREATEST FORCE, OR THAT DUE TO THE INERTIA OF THE VALVE GEAR, IS NEGLECTED.

Where flat slide valves are used it is necessary to provide a relief frame at the back of the valve so as to eliminate friction. I have about one thousand of these Cylinders in use at the present time, all of which are giving the greatest possible satisfaction to the owners.

I refer you to the following owners and builders, some of whom have had years of experience with my Cylinders:

American-Hawaiian Steamship Company, New York City, N. Y.	(2 Engines, 3,600 I.H.P.)
United States Navy Department, Washington, D. C.	(20 Engines, 200,000 I.H.P.)
Pacific Mail Steamship Company, California	(8 Engines, 30,000 I.H.P.)
Pacific Coast Steamship Company, Seattle, Washington	(5 Engines, 16,500 I.H.P.)
Merchants & Miners Transportation Co., Baltimore, Md.	(3 Engines, 9,600 I.H.P.)
Gulf Refining Company, New York City, N. Y.	(4 Engines, 12,000 I.H.P.)
Texas Oil Company, New York City, N. Y.	(2 Engines, 5,000 I.H.P.)
Standard Oil Company, New York City, N. Y.	(4 Engines, 7,200 I.H.P.)
Old Dominion Steamship Co., New York City, N. Y.	(1 Engine, 1,800 I.H.P.)
Coastwise Transportation Co., Boston, Mass.	(5 Engines, 9,000 I.H.P.)
Mallory Line Steamship Co., New York City, N. Y.	(1 Engine, 7,000 I.H.P.)
American Shipbuilding Co., Cleveland, Ohio	(12 Engines, 20,000 I.H.P.)
Kawasaki Dockyard Co., Kobe, Japan	(2 Engines, 8,000 I.H.P.)
N. Odero Company, Genoa, Italy	(4 Engines, 20,000 I.H.P.)
Vulcan Works, Stettin, Germany	(2 Engines, 5,000 I.H.P.)
United States Revenue Service, Washington, D. C.	(2 Engines, 3,600 I.H.P.)
Chesapeake Bay Line Steamer *Columbia*	(1 Engine, 3,500 I.H.P.)
Miscellaneous Small Engines	(14 Engines, 15,000 I.H.P.)
TOTAL I.H.P. IN USE 376,800.	

Many of the above engines have been in use for periods of from five to ten years, and I have numerous letters from the owners testifying to the great advantages of my patent Assistant Cylinders.

CAMDEN, N. J., *June 28, 1913.*

MR. LOVEKIN,
Chief Engineer, New York Shipbuilding Co., Camden, N. J.

DEAR SIR:

I have been Chief Engineer of the S. S. *Governor* for the past 6 years, and your Assistant Cylinders were installed on her. I found them to be a very good thing for the eccentrics, for after running 6 years the low-pressure straps had only worn $\frac{1}{16}$ of an inch, and the H.P. and M.P. the thickness of a tin shim. The engines are twin screw 5,000 I.H.P.

We ran about 370,000 miles and averaged 22 landings a month. We only had water on an eccentric once in that time, and that was when the steam pipe to the low-pressure Assistant Cylinder broke, and while we were repairing it we had to run water on it as it would not run with oil.

The only repairs we had to make to the cylinders was to put in new snap rings once a year, and I can recommend as a great saving on the valve gear and the eccentric straps.

Yours truly,

B. K. MARTLAND,
Chief-Engineer, S. S. *Congress*,
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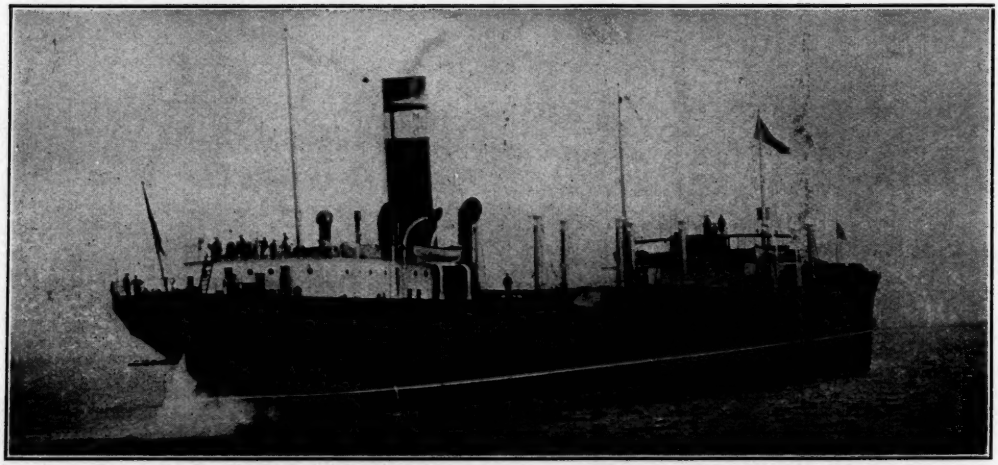

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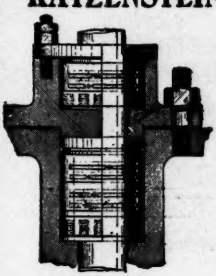

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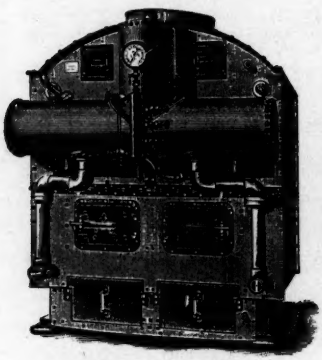

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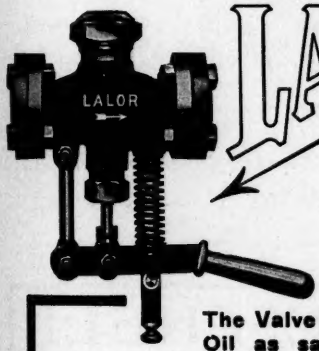

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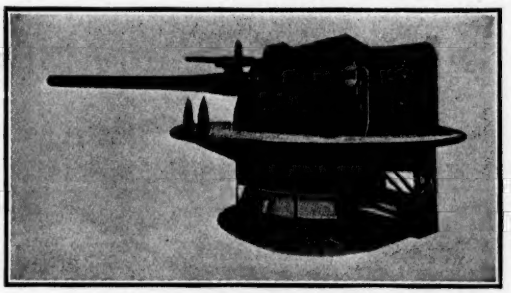

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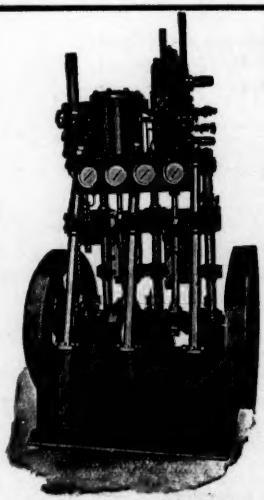

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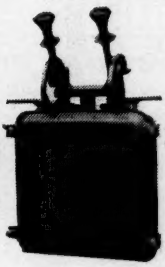

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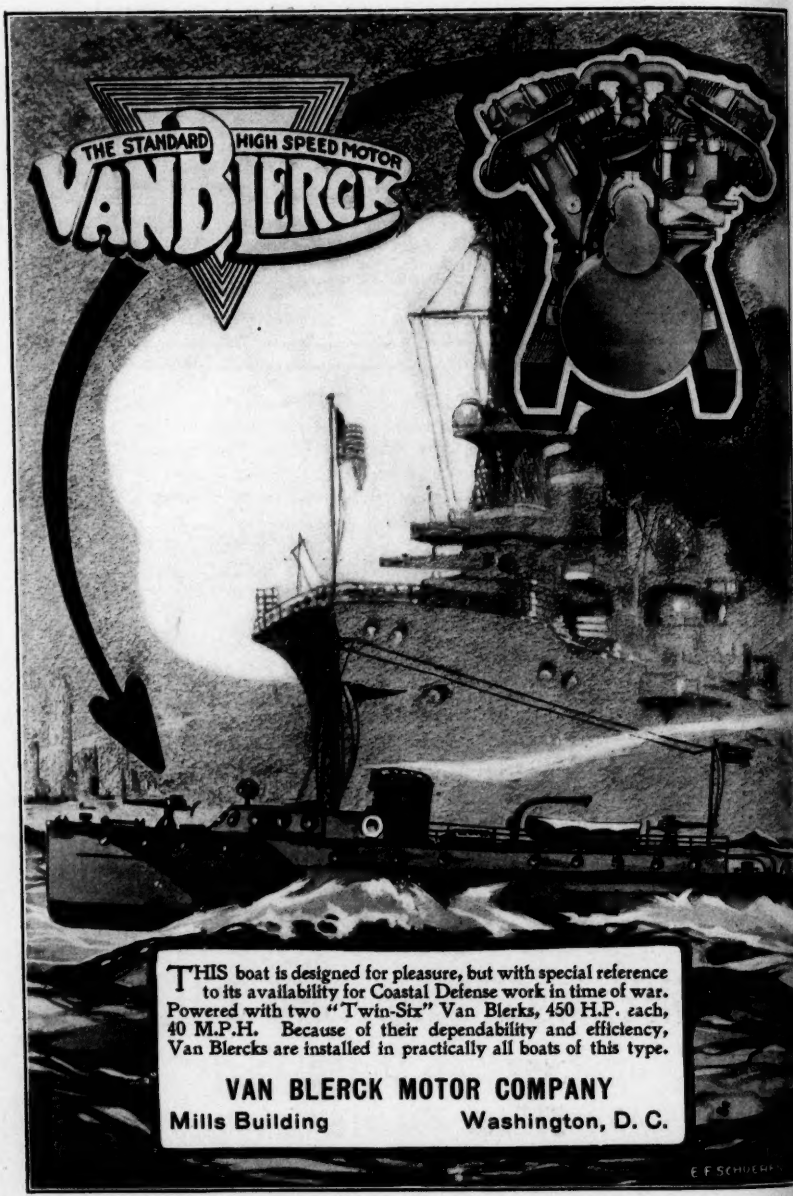
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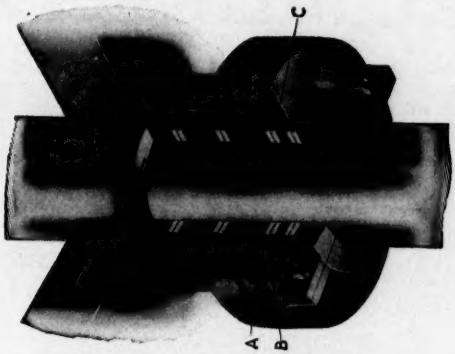

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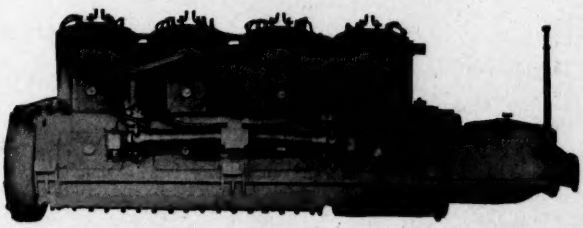

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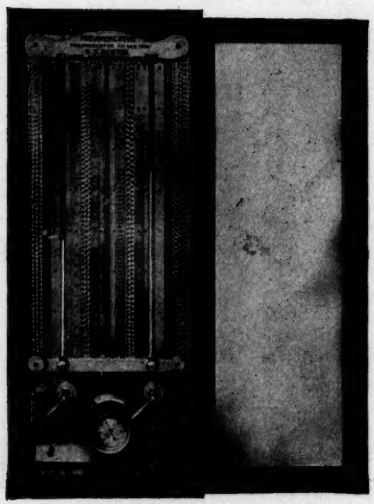

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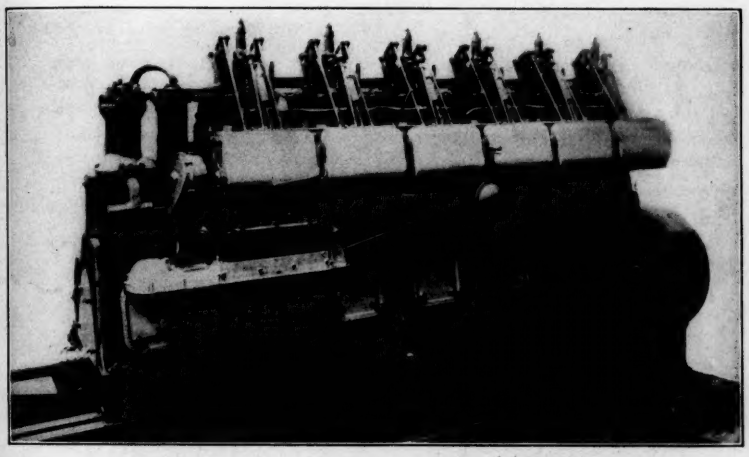

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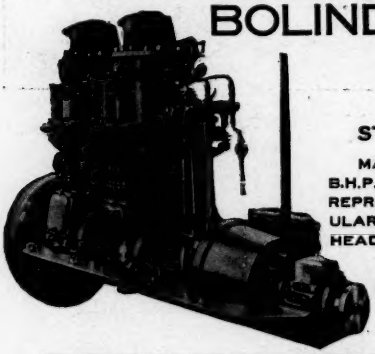

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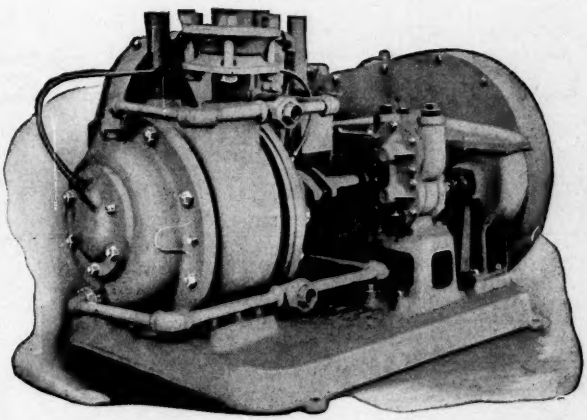

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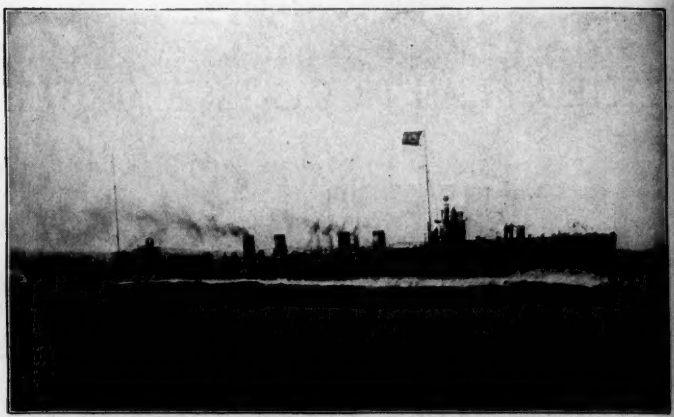

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